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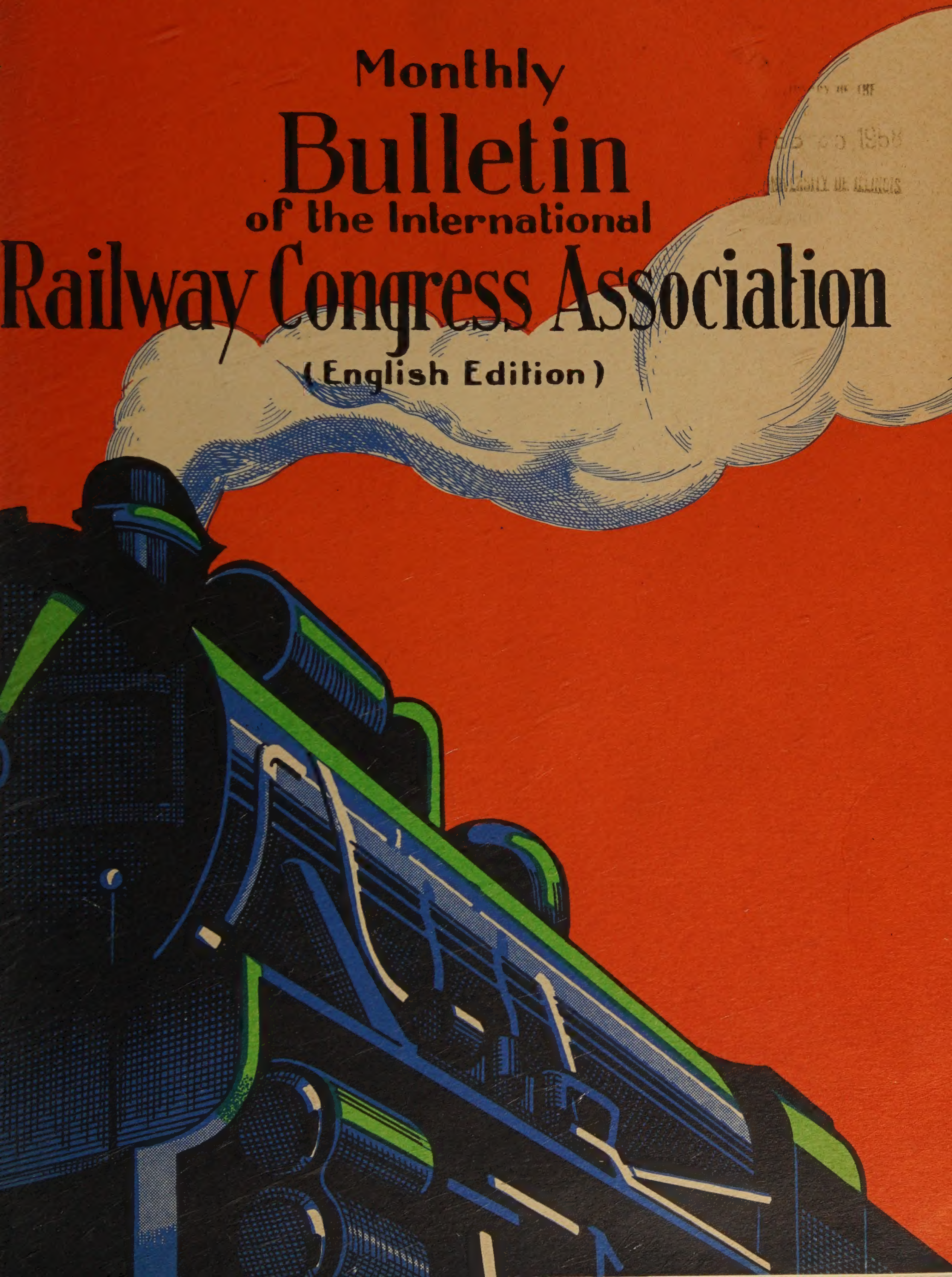
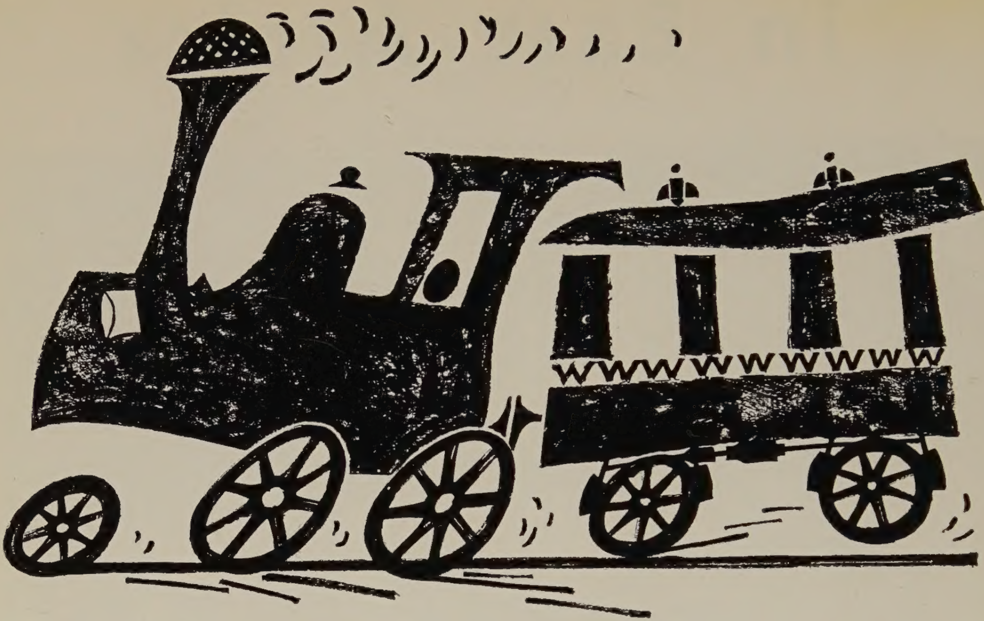
(English Edition)





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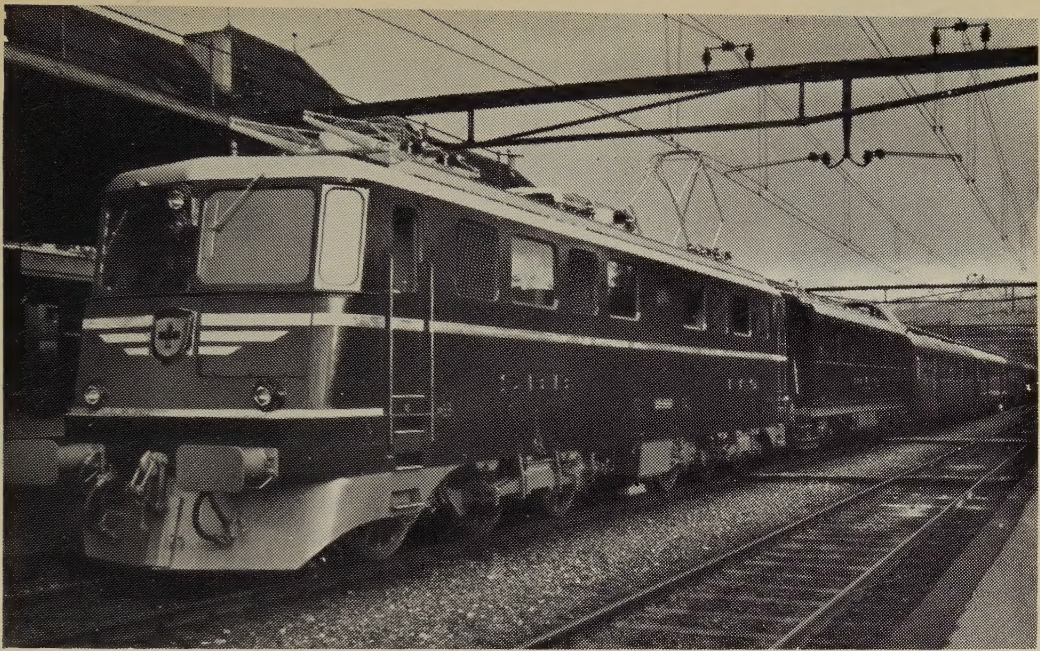
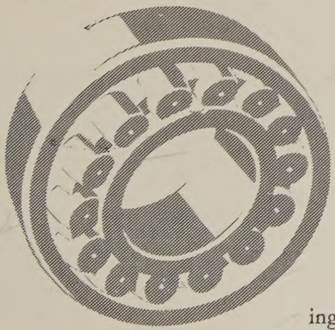


Electric locomotive, type Ae 6/6, equipped throughout with SKF bearings
Power: 6,000 h.p.
Max. speed: 75 m.p.h.
Weight in running order: 122 $^1/_2$ tons

Two 6,000 h.p. Electric Locomotives with SKF Roller Bearings



Illustrated above is one of two electric locomotives, type Ae 6/6, built by Schweizerische Lokomotiv- und Maschinenfabrik, Winterthur, for Swiss Federal Railways. These two locomotives are now in regular service on the Saint Gotthard line. The bogies — each of which has three individually driven axles —, the traction motors and the drives are fitted with SKF roller bearings.

These powerful locomotives can haul a 600-ton train on a winding track with a ruling gradient of 1 in 37 at a speed of 47 m.p.h. Even under these difficult working conditions SKF bearings are giving complete satisfaction.

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Reliability — no hot boxes
Longer mileage between overhauls
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Bulletin of the International Railway Congress Association

CONTENTS OF THE NUMBER FOR JANUARY 1958.

MONTHLY BULLETIN

OF THE

INTERNATIONAL RAILWAY CONGRESS ASSOCIATION

(ENGLISH EDITION)

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BULLETIN
OF THE
INTERNATIONAL RAILWAY CONGRESS
ASSOCIATION
(ENGLISH EDITION)

[625 .113]

Calculation of superelevation and length of transition curves for high speed working,

by J. CHAPPELLET,

Ingénieur honoraire de la Société Nationale des Chemins de fer français (Service Central de la Voie de la Région Nord).

There is, at 120 km/h (75 m. p. h.) a certain standard of riding comfort which must be maintained at speeds in excess of 120 km/h.

The degree of comfort can be defined in several ways.

We can take as a standard of comfort, the stability of a standing passenger (not supported) in the corridor of a coach over one of the two axles or bogies at the end of the coach; the destruction of this stability at first being independent of the sideways oscillation of the body caused by the action of centrifugal force.

With a passenger travelling as described above, the following two conditions are essential to ensure that he does not lose balance when the vehicle is running at a speed V km/h in a curve of radius R laid with a superelevation S :

1° The centrifugal force not compensated by superelevation should not have any obnoxious effect on the passenger through transverse force;

2° the speed of transverse rotation of the vehicle must be fairly slow if it is to be without effect on the passenger.

To say that uncompensated centrifugal force must exert no transverse force on the passenger means the same as saying that too little corresponding superelevation will not affect the passenger.

Insufficiency of superelevation I is given by the formula :

$$I = \frac{V^2 \times 1.50}{R \times 10} - S,$$

making $g = 10$. The first term of the second member being the theoretical superelevation, the second term of the second member being the superelevation S given to the curve of radius R.

We thus have to determine experimentally the amount by which the superelevation I is inadequate, by taking, in the formula given above, the value of I which is shown by speed trials to be satisfactory as regards comfort. For this purpose we will analyse some speed trials carried out in the North Region.

In the *International Railway Congress Bulletin* of October 1939, a description is given of *a speed trial 110 km/h (30 m/sec) over a curve of 2 000 m radius, laid without superelevation and having no transition curves.* The trial vehicle was a wooden coach with bogies spaced 13.68 m apart, weighing 40 t.

During this trial, we noted that the centrifugal force :

$$\frac{P}{g}\ \frac{V^2}{R} = \frac{P}{10} \times \frac{\overline{30.55}^{2}}{2\,000} = 0.0467\ P$$

developed at the tangent point had only very slight influence on a passenger as described above. In this case, we have :

Theoretical superelevation :

$$\frac{\overline{30.55}^{2} \times 1.50}{2\,000 \times 10} = 0.07$$

Insufficiency of superelevation :

$$0.07 - 0 = 0.07$$

Centrifugal force not compensated :

$$\frac{P}{10} \times \frac{\overline{30.55}^{2}}{2\,000} = 0.0467\ P$$

Speed tests in a curve of 500 m radius, laid with a superelevation equal to 0.15 m, the transition curves being 90 m long.

V = 90 km/h (25 m/sec). $g = 10$

Time taken to cover the transition curve :

$$\frac{90}{25} = 3''\,6$$

Theoretical superelevation :

$$\frac{\overline{25}^{2} \times 1.50}{500 \times 10} = 0.187\ \text{m}$$

Insufficiency of superelevation :

$$0.187 - 0.15 = 0.037$$

Variation per second from insufficient superelevation :

$$\frac{37}{3.6} = 10\ \text{mm}$$

Uncompensated centrifugal force :

$$\frac{P}{10}\left(\frac{\overline{25}^{2}}{500} - \frac{10 \times 0.15}{1.50}\right) = \frac{P}{10} \times 0.25$$

NORMAL STANDARD OF COMFORT

V = 100 km/h (28 m/sec).

Time taken to cover the transition curve :

$$\frac{90}{28} = 3''\,2$$

Theoretical superelevation :

$$\frac{\overline{28}^{2} \times 1.50}{500 \times 10} = 0.235$$

Insufficiency of superelevation :

$$0.235 - 0.15 = 0.085$$

Variation per second from insufficient superelevation :

$$\frac{85}{3.2} = 26\ \text{mm}$$

Uncompensated centrifugal force :

$$\frac{P}{10}\left(\frac{\overline{28}^{2}}{500} - \frac{0.15 \times 10}{1.50}\right) = \frac{P}{10} \times 0.568$$

ACCEPTABLE STANDARD OF COMFORT

V = 110 km/h (30.55 m/sec).

Time taken to cover the transition curve :

$$\frac{90}{30} = 3''$$

Theoretical superelevation :

$$\frac{\overline{30.55}^{2} \times 1.50}{500 \times 10} = 0.28\ \text{m}$$

Insufficiency of superelevation :

$$I = 0.28 - 0.15 = 0.13$$

Variation per second of I :

$$\frac{130}{3} = 43 \text{ mm}$$

Centrifugal force not compensated :

$$\frac{P}{10}\left(\frac{30.55^2}{500} - \frac{0.15 \times 10}{1.50}\right) = \frac{P}{10} \times 0.80$$

Having been thrown against the corridor wall, there was some hesitation before undertaking the trial at 120 km/h.

V = 120 km/h (33 m/sec).

Time taken to cover the transition curve :

$$\frac{90}{33} = 2''\,7$$

Theoretical superelevation :

$$\frac{33^2 \times 1.50}{500 \times 10} = 0.33 \text{ m}$$

Insufficiency of superelevation :

$$0.33 - 0.15 = 0.18$$

Variation per second of I :

$$\frac{180}{2''\,7} = 67 \text{ mm}$$

Centrifugal force not compensated :

$$\frac{P}{10}\left(\frac{33^2}{500} - \frac{0.15 \times 10}{1.50}\right) = \frac{P}{10} \times 1.18$$

We were thrown violently against the corridor side wall.

In addition, it may be mentioned that on the *North Region there is a permissible speed of 110 km/h in curves of 500 m radius, laid with a superelevation equal to 0.18 m, the transition curve being 90 m long.* In this case we have :

Time taken to cover the transition curve :

$$\frac{90}{3} = 3''$$

Theoretical superelevation :

$$\frac{30.55^2 \times 1.50}{500 \times 10} = 0.28$$

Insufficiency of superelevation :

$$0.28 - 0.18 = 0.10$$

Variation per second of I :

$$\frac{100}{3} = 33 \text{ mm}$$

Centrifugal force not compensated :

$$\frac{P}{10}\left(\frac{30.55^2}{500} - \frac{0.18 \times 10}{1.50}\right) = \frac{P}{10} \times 0.60$$

ACCEPTABLE STANDARD OF COMFORT

Calculation of minimum radius of a curve with a superelevation S, *below which an envisaged speed* V *cannot be achieved.*

From the formula giving insufficiency of superelevation :

$$I = \frac{V^2 \times 1.50}{R \times 10} - S,$$

we can get :

$$R = \frac{V^2 \times 1.50}{(I + S)\,10}.$$

To obtain the greatest minimum curve, it is necessary to give S the maximum permitted value, i.e. S = 0.18 m.

When V = 140/h (39 m/sec) :

I = 0.07, we find :

$$R_{min} = \frac{39^2 \times 1.50}{(0.07 + 0.18)\,10}$$

$$= \frac{1\,521 \times 1.50}{2.5} = 913 \text{ m}$$

I = 0.085 :

$$R_{min} = \frac{2\,281.5}{(0.085 + 0.18)\ 10} = 861 \text{ m}$$

I = 0.10 :

$$R_{min} = \frac{2\,281.5}{(0.10 + 0.18)\ 10} = 815 \text{ m}$$

I = 0.13 :

$$R_{min} = \frac{2\,281.5}{(0.13 + 0.18)\ 10} = 736 \text{ m}$$

I = 0.15 :

$$R_{min} = \frac{2\,281.5}{(0.15 + 0.18)\ 10} = 691 \text{ m}$$

I = 0.18 :

$$R_{min} = \frac{2\,281.5}{(0.18 + 0.18)\ 10} = 634 \text{ m}$$

When V = 150 km/h (42 m/sec) :

I = 0.07 :

$$R_{min} = \frac{\overline{42}^{2} \times 1.50}{(0.07 + 0.18)\ 10} = \frac{1\,764 \times 1.5}{2.5} = 1\,058 \text{ m}$$

I = 0.085 :

$$R_{min} = \frac{2\,646}{(0.085 + 0.18)\ 10} = 1\,000 \text{ m}$$

I = 0.10 :

$$R_{min} = \frac{2\,646}{(0.10 + 0.18)\ 10} = 945 \text{ m}$$

I = 0.13 :

$$R_{min} = \frac{2\,646}{(0.13 + 0.18)\ 10} = 853 \text{ m}$$

I = 0.15 :

$$R_{min} = \frac{2\,646}{(0.15 + 0.18)\ 10} = 802 \text{ m}$$

I = 0.18 :

$$R_{min} = \frac{2\,646}{(0.18 + 0.18)\ 10} = 735 \text{ m}$$

When V = 160 km/h (44 m/sec) :

I = 0.07 :

$$R_{min} = \frac{\overline{44}^{2} \times 1.50}{(0.07 + 0.18)\ 10} = \frac{2\,904}{2.5} = 1\,161.5 \text{ m}$$

I = 0.085 :

$$R_{min} = \frac{2\,904}{(0.085 + 0.18)\ 10} = 1\,095 \text{ m}$$

I = 0.10 :

$$R_{min} = \frac{2\,904}{(0.10 + 0.18)\ 10} = 1\,037 \text{ m}$$

I = 0.13 :

$$R_{min} = \frac{2\,904}{(0.13 + 0.18)\ 10} = 937 \text{ m}$$

I = 0.15 :

$$R_{min} = \frac{2\,904}{(0.15 + 0.18)\ 10} = 880 \text{ m}$$

I = 0.18 :

$$R_{min} = \frac{2\,904}{(0.18 + 0.18)\ 10} = 807 \text{ m}$$

Applying the formula :

$$I = \frac{V^2 \times 1.50}{R \times 10} - S$$

to the calculation of superelevation S *for* $V = 120$ *km/h.*

This formula can be expressed :

$$S = \frac{V^2 \times 1.50}{R \times 10} - I.$$

When I = 0.07 :

R = 1 000, we have :

$$S = \frac{\overline{33}^{2} \times 1.50}{1\,000 \times 10} - 0.07 = 0.093$$

R = 800,

$$S = \frac{1\,633.5}{800 \times 10} - 0.07 = 0.134$$

When I = 0.085 :

R = 1 000, we have :

$$S = \frac{1\,633.5}{1\,000 \times 10} - 0.085 = 0.078$$

R = 800,

$$S = \frac{1\,633.5}{800 \times 10} - 0.085 = 0.119$$

When I = 0.10 :

R = 1 000, we have :

$$S = \frac{1\,633.5}{1\,000 \times 10} - 0.10 = 0.06$$

R = 800,

$$S = \frac{1\,633.5}{800 \times 10} - 0.10 = 0.104$$

When I = 0.13 :

R = 1 000, we have :

$$S = \frac{1\,633.5}{1\,000 \times 10} - 0.13 = 0.033$$

R = 800 :

$$S = \frac{1\,633.5}{800 \times 10} - 0.13 = 0.07$$

When I = 0.15 :

R = 1 000, we have :

$$S = \frac{\overline{33}^2 \times 1.50}{1\,000 \times 10} - 0.15 = 0.01$$

R = 800 :

$$S = \frac{\overline{33}^2 \times 1.50}{800 \times 10} - 0.15 = 0.05$$

When I = 0.18 :

R = 1 000, we have :

$$S = \frac{\overline{33}^2 \times 1.50}{1\,000 \times 10} - 0.18 = -0.02$$

R = 800 :

$$S = \frac{\overline{33}^2 \times 1.50}{1\,000 \times 10} - 0.18 = 0.02$$

Amounts of superelevation provided on networks when V = 120 km/h :

On the Nord Railway :

when R = 1 000,

$$S = \frac{120}{1\,000} = 0.12 \text{ and } S = \frac{100}{1\,000} = 0.10$$

when R = 800,

$$S = \frac{120}{800} = 0.15 \text{ and } S = \frac{100}{800} = 0.125$$

On the Est Railway :

when R = 1 000,

$$S = \frac{12\,V^2}{R}, \text{ with } V = 100 \text{ km},$$

$$S = \frac{12 \times \overline{100}^2}{1\,000} = 120 \text{ mm}$$

when R = 800,

$$S = \frac{12 \times \overline{100}^2}{800} = 150 \text{ mm}$$

On the P.L.M. Railway :

when R = 1 000,

$$S = \frac{2}{3} \times \frac{120}{1\,000} = 0.08$$

when R = 800,

$$S = \frac{2}{3} \times \frac{120}{800} = 0.10$$

Comparison between superelevations calculated according to the formula :

$$S = \frac{V^2 \times 1.50}{R \times 10} - I$$

and amounts of superelevation allowed by the Railway companies.

The formula :

$$S = \frac{V^2 \times 1.50}{R \times 10} - I$$

gives superelevations which are comparable with those allowed by the former Nord and Est Companies, when V = 120 km/h, when I = 0.07 and allowed by the P.L.M. when I = 0.085.

Consequently, these insufficiencies of superelevation guarantee the standards of comfort which were achieved by the Railway Companies, the variations per second in the superelevation insufficiency being given by the length of transition generally equal to 90 m.

CONCLUSIONS

It can be accepted without argument that I = 0.085 or at the most 0.10, since this insufficiency of superelevation corresponds to operation at 110 km/h as permitted by the Nord Region, in curves of 500 m radius laid with 0.18 m superelevation and having transition curves 90 m long which as we have seen agrees with operation at 120 km/h on curves of 1 000 m radius with superelevation of 0.06 m.

In our opinion, however, this is a limit which must not be exceeded.

In fact, is it possible to operate in good conditions, without discomfort, without excessive shock, in a curve of 1 000 m radius, either with a superelevation equal to 0.03 or to 0.01, which is in fact without superelevation, or even, which would be more surprising, with a counter slope of 0.02?

Experience says not, because it is necessary to take account of defects in track maintenance, which are inevitable.

Certain engineers say it is possible, if sufficient running time in the transition curve leads the passenger when standing, imperceptibly to the changed position caused by the uncompensated centrifugal force.

It may be noted that for a standing passenger, *who is not expecting anything untoward, 3 sec to 4 sec is almost instantaneous* and this represents transition curves of lengths varying between 130 m and 180 m, which are very difficult to insert between the curve and the straight.

It would be necessary at least to double these lengths, and even *6 sec pass very quickly.*

However, even if we were to achieve sufficiently long curves, experience shows that running in the circular curve would have serious faults because of the inevitable defects in maintenance.

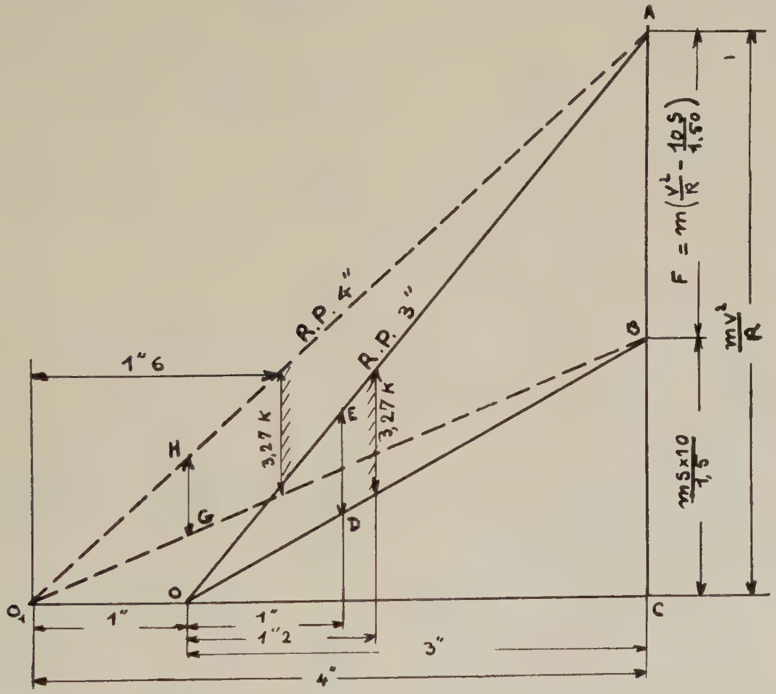
Even if it were possible, an idea immediately comes to mind — why have engineers of our generation and previous generations bothered about superelevation — often very inconvenient in certain layouts, and difficult to achieve because of widely differing speeds. It would be necessary to believe that all these engineers have been lacking in technical ability and common sense in a matter where experience alone would provide a solution which, if not perfect, would be the best possible.

It is obvious that the solution of superelevation for very high speed running is in the construction of carriages, the bodies of which would automatically and progressively take up an inclination I' so that the total of this inclination, the insufficiency of superelevation and the superelevation provided would be equal to the theoretical superelevation; that is, there would be : I' + I + S = theoretical superelevation.

Thus, for a theoretical superelevation equal to 0.33 m, an applied superelevation equal to 0.15 m, a permissible insufficiency I = 0.08, the body would have to take an inclination, equal to 0.33 — (0.08 + 0.15) = 0.10. The length of the transition curves would be governed by the permissible variation in superelevation : in our example, this length would be equal to :

$$\frac{150}{1.5} = 100 \text{ m}$$

for a variation of superelevation equal to 1.5 mm.



Calculation of the length of a parabolic curve.

The length of a parabolic transition curve can be determined by using the results of speed trials.

Taking the case of a curve of 1 000 m radius, covered at a speed of 40 m/sec, we get :

Theoretical superelevation :

$$\frac{\overline{40}^2 \times 1.50}{1\,000 \times 10} = 0.24.$$

If a superelevation equal to 0.14 is provided, the insufficiency of superelevation is equal to 0.24 — 0.14 = 0.10.

We will take a superelevation variation per second equal to 33 mm which will give a length of transition curve L :

$$L = \frac{100}{33} \times 40 = 3''3 \times 40 = 132 \text{ m},$$

or thirteen to fourteen 10 m stages.

Moreover, the length L should not give a superelevation variation i greater than the permissible value.

In the foregoing example, we would have :

$$i = \frac{140 \text{ mm}}{140} = 1 \text{ mm}.$$

Considérations on increasing the length of a transition curve.

When it is necessary to increase the length of a transition curve to reduce thrust caused by insufficient superelevation, the investigation into lengthening the transition curve can be approached in the following manner :

Firstly, insufficiency of superelevation corresponds to an excess of uncompensated centrifugal force.

Consequently, the uncompensated centrifugal force being given by the formula :

$$m\left(\frac{V^2}{R} - \frac{S \times 10}{1.50}\right) = \frac{V^2}{R} - \frac{S \times 10}{1.50},$$

with $m = 1$ and $g = 10$, we can draw a diagram of the variation per second in centrifugal force, carrying the abscissae as time t'' and the ordinates as the centrifugal force :

$$\frac{m\ V^2}{R} = AC,$$

and the centrifugal force compensated by superelevation :

$$BC = \frac{m \times S \times 10}{1.50},$$

making $m = 1$. AB is the uncompensated centrifugal force F. Diagram OAB is that of the variation per second when the transition curve is covered in 3 sec; diagram O_1AB is that of the variation per second when the transition curve is covered in 4 sec.

After one second the diagram OAB shows a force :

$$ED = \frac{F \times 1''}{3} = \frac{F}{3}.$$

After one second the diagram O_1AB shows a force

$$GH = \frac{F \times 1''}{4} = \frac{F}{4}.$$

The thrust from the uncompensated centrifugal force is obviously less in the transition curve covered in four seconds than in the curve covered in three seconds.

The reduction is equal to :

$$\frac{F}{3} - \frac{F}{4} = \frac{F}{12}.$$

For an uncompensated centrifugal force equal to :

$$F = \frac{P}{g} \times 1.18,$$

which corresponds to an insufficiency of superelevation of 180 mm, the transmitted difference received by a passenger weighing 70 kg, after 1 second, is equal to :

$$\frac{1}{12} \times \frac{70}{10} \times 1.18 = 0.689 \text{ kg}.$$

This is negligible.

Let us calculate the uncompensated centrifugal forces at various times t and the thrusts received by a passenger weighing 70 kg.

Transition curve covered in 4'' :

Time	0''	1''	2''	3''	4''
Forces	0	$\frac{F}{4}$	$\frac{F}{4} \times 2'' = \frac{F}{2}$	$\frac{F}{4} \times 3$	F
Thrust	0	2.06 kg	4.13 kg	6.19 kg	8.26 kg

Transition curve covered in 3'' :

Time	0''	1''	2''	3''
Forces	0	$\frac{F}{3} \times 1'' = \frac{F}{3}$	$\frac{F}{3} \times 2'' = \frac{2F}{3}$	F
Thrust	0	2.75 kg	5.51 kg	8.26 kg

In addition, if we recall that an uncompensated centrifugal force equal to :

$$\frac{P}{10} \times 0.467,$$

or for a passenger weighing 70 kg a thrust equal to :

$$\frac{70}{10} \times 0.467 = 3.269 \text{ kg},$$

has only slight effect on our standing passenger, the diagrams show us that in the transition curve covered in 4 sec, the passenger does not feel the centrifugal force; that is a force greater than 3.269 kg, until 1.6 sec; whilst when the curve is covered in 3 sec he is subject to a force greater than 3.269 kg after 1.2 sec, and thus more quickly, but very little indeed. Finally, all this happens as if it were instantaneous, 2.4 sec for one, 1.8 sec for the other.

The lengthening of the curve from 3 sec to 4 sec thus does not noticeably improve the run in to the insufficiency of superelevation.

A similar investigation will have to be made each time it is proposed to lengthen a curve : this will allow careful assessment of the need for increasing superelevation.

[625 .244 (4)]

The new refrigerating wagons for European refrigerated traffic,

by Dipl. Eng. Ernst SCHRÖDER, Minden (Westphalia).

(*Eisenbahntechnische Rundschau*, No. 4, April 1957.)

In Europe as in other continents, the transport of perishables in refrigerator cars has essentially increased during recent years and is expected to currently gain more importance. Such transports, particularly when moved over great distances, confront the railways with a number of car design problems which are discussed by the author in the light of his long years of experience in wagon building.

The bases of refrigerated railway transport from the mechanical and constructional points of view.

The increasing demand for refrigerated transport [1] has encouraged the railways to take all possible steps, even from the mechanical point of view, to perfect to the utmost possible the design of refrigerating wagons, their insulation, their pre-refrigeration, their refrigerants, and the replacement of refrigerants. In addition, it was necessary to design arrangements allowing of speedy loading and unloading, and above all to study suitable packings to protect refrigerated goods from damage, assure proper ventilation, and rapid stowing. This is the only way to reduce the time lag in the turn round of refrigerated wagons, which is not yet satisfactory. The increase in the number of fast parcels trains which is moreover necessary quite apart from this, is not in itself sufficient. It is only if the whole chain of refrigeration, i.e. everything which affects the transport of perishable goods from the producer to the consumer, is considerably improved both as regards quality and time, that there is any hope of retaining this increasing refrigerated traffic on the railway.

The use of low temperatures, so far superior to any other method of conservation, has dictated the whole evolution of the design of refrigerated wagons. The sources of cold used today are : ordinary ice, ordinary ice mechanically pulverised into snow, a mixture of ice and salt, dry ice, and a mixture of water and dry ice.

To obtain temperatures of + 3 to + 5° C. in the wagon body, ordinary ice is used, it being simple and cheap to cool the wagons in this way.

In the case of goods which require temperatures down to — 10° C., the simplest and cheapest way is to add salt to ordinary ice. The cooling temperature obtainable in the ice compartment depends on the amount of salt in the mixture. The most favourable proportion is about 22.4 % salt and 77.6 % ice. But if the use of a mixture of ice and salt is simple and cheap as regards obtaining temperatures below zero, it has serious drawbacks as regards the wagon itself, as the steel parts coming into contact with the salt water rust very badly. Track installations and bridges are also affected.

The use of dry ice is better for the wagon, and cleaner. This is solidified carbonic acid which changes directly from the solid state into gas without liquifying and

[1] A detailed report on this subject will be found in the article : « Entwicklung und Gestaltung des europäischen Kühlverkehrs » (Development and organisation of European refrigerated transport) by Dr. W. PISCHEL, Nos. 8 and 9, 1956, of « E. T. R. ».

without leaving any residue. According to the amount of dry ice used — which as a temperature of — 78.9° C. itself — in the wagon, it is easy to obtain temperatures of — 20 to — 25° C. inside the wagon. The emission of cold by the dry ice can be slowed down by packing it in corrugated cardboard, so as to observe the temperature prescribed for the goods and reduce the consumption of dry ice to the minimum. This has the advantage over ordinary ice that only a given amount is used as determined by calculation of the temperature required, since the dry ice turns into gas completely. With ordinary ice, it is always necessary to provide a greater quantity than the amount actually needed according to the calculation of the amount of cold produced by this ice, in order to have a sufficiently large cooling surface for the size of wagon.

At a temperature of 0° C., the amount of cold available in a kg of dry ice is 152 kcal/kg, compared with 80 kcal/kg in the case of ordinary ice. The two quantities of cold have therefore a ratio of 1.9 to 1 to each other. Unfortunately, the cost of dry ice at the present time is ten times as great as that of ordinary ice, which imposes certain limitations upon its use. It is possible that an increase in the consumption of dry ice used in refrigerated transport will lead to a reduction in its price owing to more rational mass production methods. From the point of view of the maintenance of refrigerating wagons, the railway as well as the users would be delighted.

In designing an up-to-date refrigerated wagon, the results obtained in service with the types of wagons formerly used naturally play an important part. But in addition, the results of precise measurements of temperature, taken at different points in the wagon, both when stationary and during running, are also of capital importance. The German Federal Railways for some years have had an up-to-date measuring wagon, which is particularly well equipped, and makes it possible to take all the temperature readings required, not only in one wagon but throughout a rake of a certain length. In this way, it is possible to get a clear idea of the value of the insulating materials used, as well as of any undesirable paths of loss. By means of this wagon not only has the refrigerated transport of the Deutsche Bundesbahn been studied, but also that of the Transthermos Company. In this way it has been possible to ascertain the continuous variation in temperature at all desired points in a refrigerating wagon from the time it is loaded until it is unloaded. A record is also made of the quantity of cooling material used during the journey.

When studying the basic design of a series of refrigerated wagons, it is essential to decide first of all the purpose for which they are going to be used. The kind of goods to be carried in cold storage, the temperature required inside the wagon, and the length of the run, the question of stocking up again with ice, and the coefficient of heat transmission specified must all be known with certainty before the studies are begun.

Designing modern types of refrigerating wagons.

Owing to the international importance of refrigerated transport, the standardisation of refrigerator cars has been thoroughly gone into in recent years by the International Railway Union (U. I. C.) and the Research and Tests Office (O. R. E.). All the important characteristics of general interest have been defined in regulations and recommendations for the European Railways. From the point of view of the coefficients of heat transmission obtainable in refrigerated wagons (K in kcal/m² h °C.) two standard types have been designed :

type 1, with average insulation :

$K \leqslant 0.6$ kcal/m² h °C., and

type 2, with heavy insulation :

$K \leqslant 0.35$ kcal/m² h °C.,

which latter can also be used for goods which have to be carried at very low temperatures.

During the elaboration of the U. I. C.

and O. R. E. regulations concerning refrigerated wagons, three series of such wagons have been built in Europe :

1) in 1951, after the foundation of Interfrigo, 525 refrigerated wagons were built for this organisation, 350 to the British gauge and 175 to the continental gauge;

2) in 1955, Interfrigo ordered a further 350 wagons conforming to the U. I. C. regulations for type 1 above, with average insulation K $\leqslant$ 0.6 kcal/m² h °C;

3) during the same period, the D. B. converted into all-metal wagons according to the latest methods its universal type refrigerated wagons usable for all types of goods, and built some 500 such wagons. These corresponded to the U. I. C., type 2, K $\leqslant$ 0.35 kcal/ m² h °C.

These series built in Europe during the last five years on the one hand include all the improvements made in refrigerated wagons, and on the other hand, show the results of the U. I. C. standardisation. In describing these types, the most important characteristics of which are shown in the table, we will make particular mention of any innovations in the construction of refrigerated wagons.

1951 Interfrigo refrigerated wagons.

Interfrigo had its 525 four wheeled refrigerated wagons built by French, Swiss and Belgian firms (figs 1 and 2). The frames of these wagons are made of welded rolled sections; they are supported by a double ring suspension with a 1 400 mm long spring with seven plates (120 × 16 mm) on axles mounted on roller bearings boxes with SKF type spherical tapered bearings. The framework of the wooden wagon body is made of rolled steel sections. The uprights of the sides are standard 100 U sections, the corner pillars 90 × 90 × 8 equal angle irons and the uprights of the end walls by 100 × 65 × 11 angle irons reinforced by a flat welded to them. The side wall between the corner pillar and the first upright of the side is covered with 4 mm plate. This plate takes the place of the side bracing and is intended to transmit buffing shocks from the body to the frame. The other panels of the sides and the doors are covered in the usual way with 15 mm pine boards.

The inside walls are made of hard-wood fibre board; they are screwed on the wooden rails of the sides, so arranged in the sides as to give rise to as few points of loss as possible. A sheet of galvanised corrugated steel is screwed onto the fibre board inside the wagon; it is 1.5 mm thick, with 20 mm deep corrugations 50 mm long. The corrugated sheet is intended to protect the sides from mechanical damage from the load. In addition, it encourages good circulation of the air inside the wagon even when goods are stacked close to the sides.

The wood floor rests on the oak cross bearers bolted to the frame. A false floor of 1.5 mm steel plate rests on the longitudinals of the frame and supports of the body to carry the floor insulation. The wood floor is covered with a granilastic rubber-based floor covering, applied by trowel, 10 mm thick to assure its imperviousness. This covering is in addition sufficiently elastic, even in service, so that it does not crack even if the body is slightly deformed in service, which it is not always possible to avoid. As, moreover, its coefficient of thermal conductivity is low, the coefficient of heat transmission of the wagon is improved.

At each end of the wagon there is a container for the ice. To catch the water dripping from the ice container, the floor is lowered at both ends to form a trough to catch this water (fig. 1). The water from the melting ice is taken towards the exterior by two drains acting as siphons. These devices for getting rid of the water are designed in such a way that hot air from outside cannot penetrate into the wagon.

The roof of the wagon is covered with 1.5 mm thick steel sheet welded to the metal roof sticks. Inside, the roof is covered with 15 mm thick pine boards (fig. 2, on left). Two lines of Flettner rotors are fitted on the roof. These are double rotors in which the two superimposed rotors are set out of line by 90°. This arrangement facilitates the starting of the rotors, even

at low air speeds. These rotors drive the fans for stirring up the air inside the wagon by means of a shaft through the roof. To improve their efficiency, these fans have guide plates made of galvanised sheet. They draw the air from the middle walls of the wagon with a fixed front wall separating it from the loading compartment (fig. 1). The ice container is covered with expanded metal fixed about 50 mm away from the walls (fig. 2, on left). The air sent over the ice container by the fans

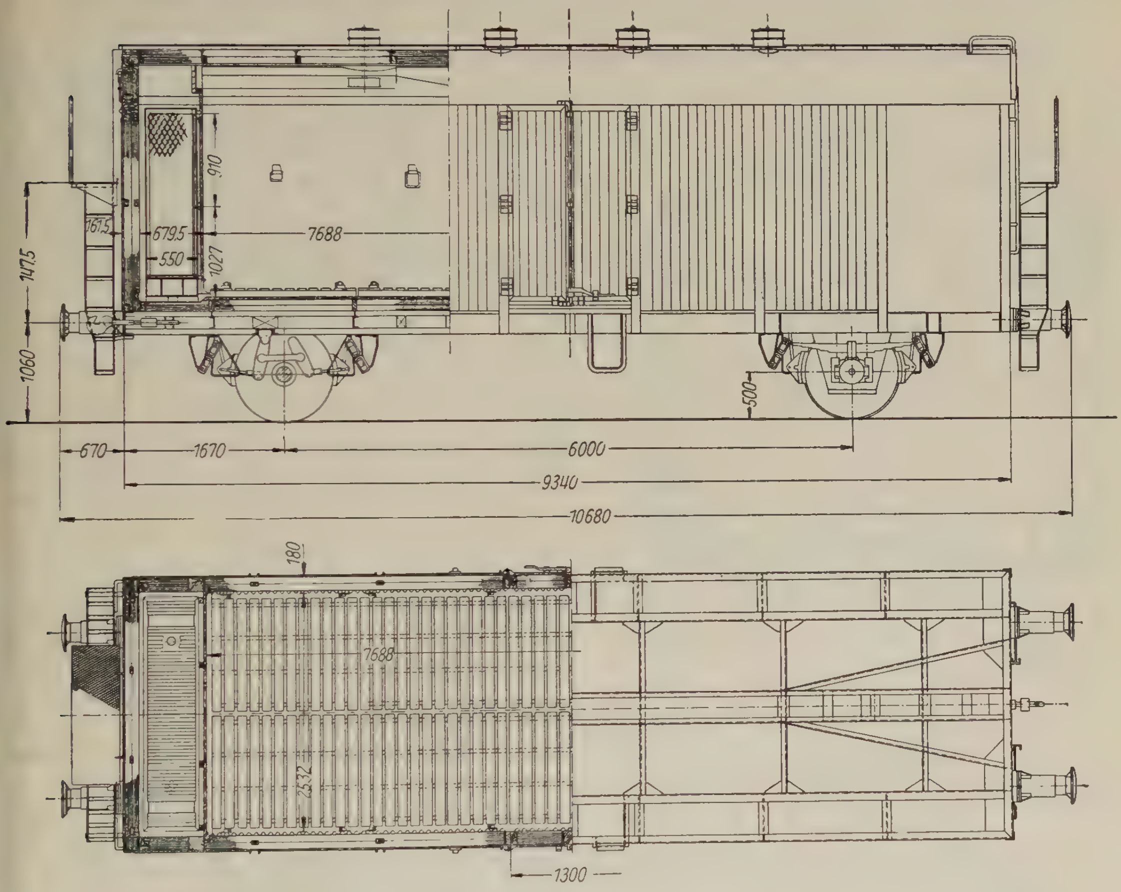


Fig. 1. — Interfrigo refrigerated wagon with two axles. Above: elevation and section. Below: frame and section of body. Scale 1 : 80.

of the wagon upwards towards the roof and send it to the ends of the wagon where it is cooled by contact with the ice in the ice containers.

The 2.4 m³ ice container is fitted along each end of the wagon over the whole width. It is formed by the end and side is thus able to flow all over the ice, so that this gives the large area of cooling surface required to obtain a low temperature inside the wagon. The ice is placed on a grill in the container about 20 cm above the floor. The fixed front of the container starts at about the same level

above the floor and separates the interior of the wagon from the ice container. In the upper part, all along the roof, ducts carry the air over the ice from openings in the partition. The cooled air returns to the interior of the wagon by passing through the grill under the ice. A double door in the partition makes it possible to insert the ice from inside the wagon and also to remove the ice from the container and clean it out.

The floor is covered with 85 mm thick duck-boarding. The space under the cross slats of the boarding longitudinally allow the air coming out of the ice container to flow all over the wagon under the load. Duck-boarding with a sufficiently large space beneath is absolutely essential for satisfactory air circulation. It is hinged to the bottom of the side walls; it is lifted when the wagons are cleaned out.

On each side of the wagon there is a double door for loading 1 300 mm wide and 2 000 mm high. Hollow rubber mouldings ensure the tightness of the doors which are closed by a cam shaft arrangement. On the upper part of each end wall, under the roof, there is a rectangular trapdoor for loading the ice into the ice container. A loading platform on each end reached from either side by steps enables the staff to put the ice into the container without any other handling gear.

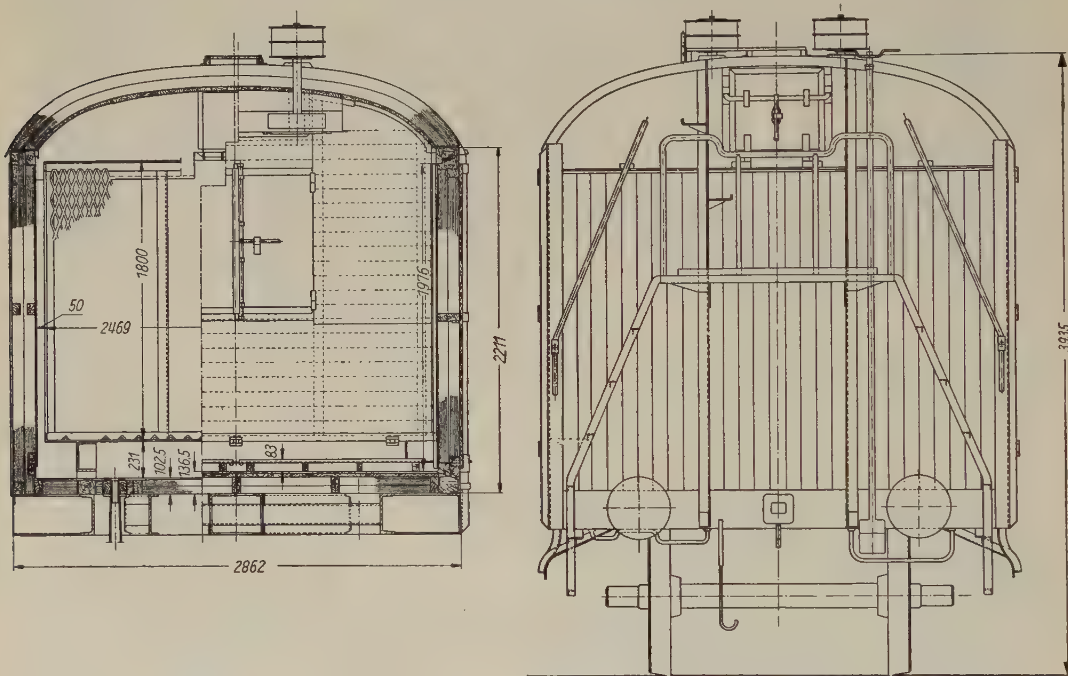


Fig. 2. — Interfrigo refrigerated wagon with two axles. Left: section of body; right: end view. Scale 1 : 40.

The floor and walls of the wagon are insulated with Onazote and the roof with Isoflex. Onazote is a hard foam rubber with a high specific weight of 64 kg/m³ and a coefficient of heat conductivity of about 0.03 kcal/m² h °C. The insulating isoflex sheets are plysheets made up of sheets of plastic materials crossed and stuck together hot under pressure. Use is made

of sheets of cellulose acetate as well as polyvinol chlorate. The specific weight of isoflex is 12 to 14 kg/m³, and its coefficient of heat conductivity, $\lambda = 0.04$ kcal/m² h °C. In the walls, roof and floor, the insulation is 140 mm thick. The coefficient of heat transmission of the wagon, determined by the heating method laid down by the I. R. U. (U. I. C.) amounts, when new, to about 0.55 kcal/m² h °C. Great importance has also been attached to the tightness of the body which is essential in order to obtain a good coefficient of heat transmission. To check the tightness, air is sent into the wagon through the water drains and the amount lost through defects in air-tightness are measured. The wagon passed these tests satisfactorily.

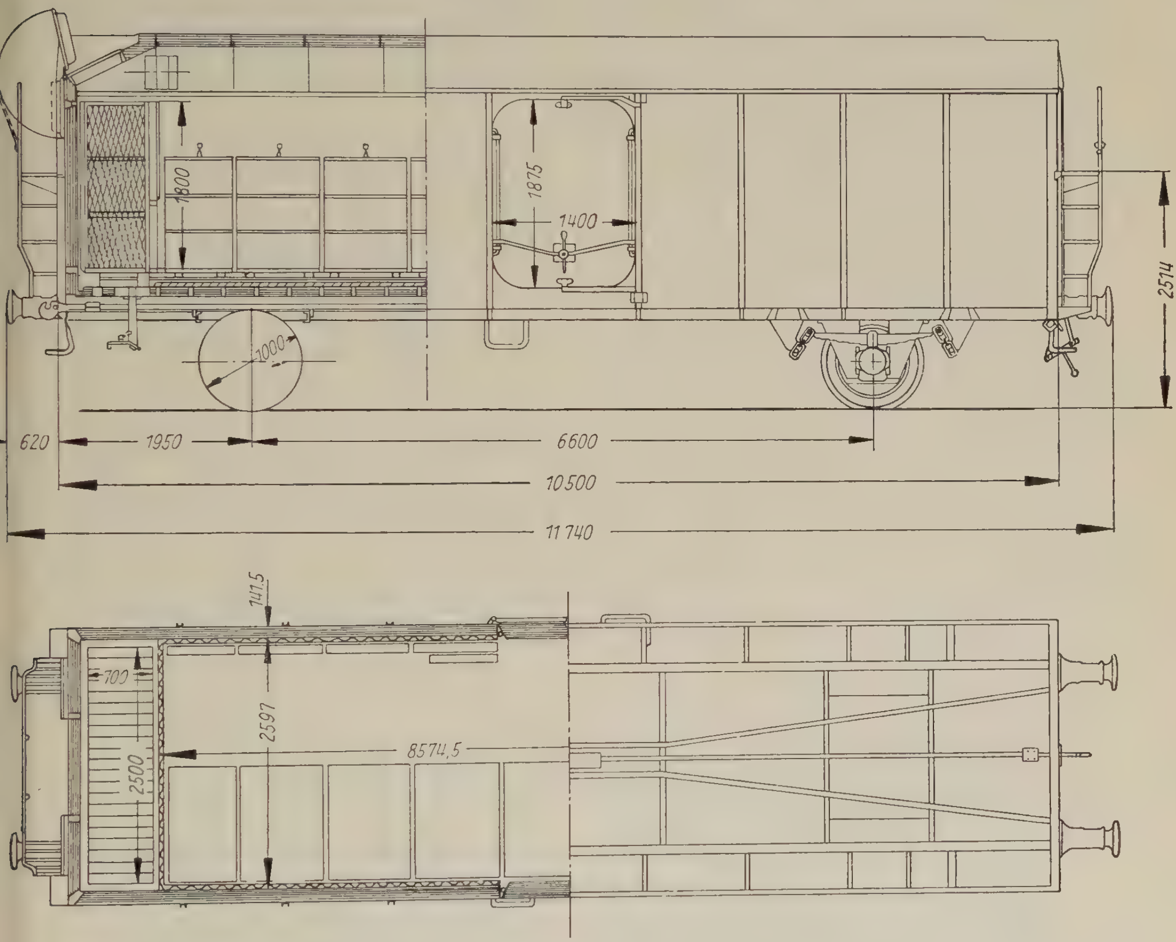


Fig. 3. — Interfrigo refrigerated wagon with two axles, U. I. C. pattern, type 1. Above: elevation and section; below: frame and section of body. Scale 1 : 80.

The tare of the wagon, 17 500 kg, seems rather high for a four wheeled wagon of about 10.5 m length over buffers and

TABLE : **Essential characteristics of the new types of refrigerated wagons.**

1	2	3	4
Characteristics of the wagons	Interfrigo wagon of 1951	Interfrigo wagon U.I.C. type 1 of 1955	Deutsche Bundesbahn refrigerated wagons U.I.C. pattern, type 2
1. Tare . . . . t	17.8	about 16.5	15.8
2. Weight loaded in regular service on lines of categories . . . . A : t	15.0	15.5	16.0
B : t		19.5	20.0
C : t		19.5	20.0
Maximum weight loaded in express service . . . . t	15.0	19.5	20.0
3. Available length for load mm	7 688.0	8 574.5	8 527 with / 9 665 without ice containers
4. Available width for load mm	2 532.0	2 597.0	2 297
5. Area available for load. m²	19.5	22.0	19.5 with / 22.2 without ice containers
6. Volume available for load . . . . m³	40.0	40.0	37.0 with / 42.2 without ice containers
7. Content of the two ice containers (total) . . . m³	4.5	6.4	4.5 ordinary ice; 1.6 dry ice
kg	2 500.0	3 500.0	2 500.0 ordinary ice; 850.0 dry ice
8. Maximum speed . . . km/h	100.0	100.0	100

The standard categories of lines include lines with permissible axle loads of : A : up to 16 t; B : above 16 to 18 t; C above 18 up to 20 t.

140 mm thickness of insulation. It is due to the extremely robust construction of the body and the use of a relatively heavy insulation in the floor and walls. The main characteristics of the refrigerated wagon are given in column 2 of the table.

Interfrigo U. I. C. pattern, type 1, 1955 wagon.

Owing to the development of refrigerated transport, in 1955, Interfrigo decided to increase its stock of wagons by building 350 new four wheeled refrigerated wagons. Since the older wagons were built, the U. I. C. refrigerated wagon with average insulation (type 1) had been designed, so that Interfrigo decided to order wagons of this type (fig. 3 to 5). This decision was justified by the fact that owing to experience acquired in the meanwhile, the new type had many interesting improvements over the old type. The body of the wagon, now built entirely of metal, has

hermetically welded sheet walls on the outside and corrugated galvanised sheet walls on the inside. The doors, as a result of experience acquired in America, are single doors for the first time. The trap-door for loading the ice is now arranged obliquely in the roof along the end wall, to facilitate the use of machines for icing the wagons without special elbow connections.

As an experiment some of the wagons were fitted with electric fans fed during running from a generator driven by the axle. When the vehicle is stationary, for example when carrying out its pre-refrigeration, it is possible to feed these fans from the local mains. All the other wagons are fitted with Flettner rotors and fans of the standard pattern.

The U. I. C. has adopted for its two types of refrigerated wagons : type 1 with average insulation and type 2 with heavy insulation, a standard frame, that of type 2 with which the D. B. has already built several hundreds of wagons (fig. 3). The running gear of these new Interfrigo wagons, fitted with axles with roller bearings, long suspension by laminated springs and with brake gear, corresponds to those of the first Interfrigo wagons of 1951.

The outside framework of the body, entirely welded (fig. 3) is made of 1.5 mm sheet and rolled sections which are used for the body pillars, the end stanchions, the corner pillars and the door frames. In these refrigerated wagons with average insulation, the body sheets are placed behind the pillars of the side or end walls, whereas in the type 2 with increased insulation, the wall pillars are behind the body covering sheets; only in this way was it possible for a given width of body and as big a loading area as possible to be obtained and to fit in the thicker insulation used with this type (250 mm). For the sake of exterior cleanliness, to facilitate cleaning and give a better appearance, preference should be given to the second type with smooth outer walls.

The interior covering of the walls consists of corrugated galvanised sheet with 20 mm corrugations, the bottom of which rests on a wooden frame screwed to the oak floor cross bearers (fig. 4). The interior corrugated sheet lining is only connected to the body at the top, at the level of the upper frame. The meat hooks are fastened to the side walls of corrugated sheet. Part of the wagons are fitted with 300 fixed meat hooks. The walls of corrugated sheet together with the duck-boarding used for the floor, assure good circulation of air throughout the wagon, an essential condition for proper refrigeration of the load.

The floor of the wagon consists of 32 mm thick pine boards resting on oak beams fixed to the cross members of the frame. The false floor of 1.5 mm steel sheet rests on the frame, forming the body covering at the bottom and carrying the floor insulation (fig. 3). Inside, as on the first 525 Interfrigo wagons, the wooden floor is covered with an 8 mm thick layer of granilastic. The wagons being built in Germany for Interfrigo have a layer of Semtex instead of Granilastic, which is also a rubber-based floor covering laid by trowel.

The outer covering of the roof is 1.5 mm corrugated sheet. This is welded onto the steel roof sticks which rest on the roof beams (fig. 5). Inside the roof is covered with an 0.8 mm thick galvanised sheet. These sheets are screwed into the wooden hoop sticks fixed to the steel outer trusses by means of wooden packing pieces.

At each end of the wagon, there is a 3.2 m^3 ice container to take the ice used for cooling the load. The container, which is 700 mm deep, is formed by the end wall of the wagon, the two side walls and a fixed partition in front. The ice is placed in a special box over a grill in this container, about 200 mm above the floor of the wagon. The sides of the ice box are covered with expanded metal and are about 50 mm away from the walls. This expanded metal covering of the ice box increases the effective surface of the ice in the tank by allowing air to circulate around the block. In the front partition, there is a double leaf door for emptying the container so that it can be cleaned out. The

container can also be filled through this door from inside the wagon.

Under the ice container, the floor of the wagon is slightly hollowed out, to collect the water from the melting ice (fig. 3, above). The water runs out of the wagon by means of two drainage devices at each

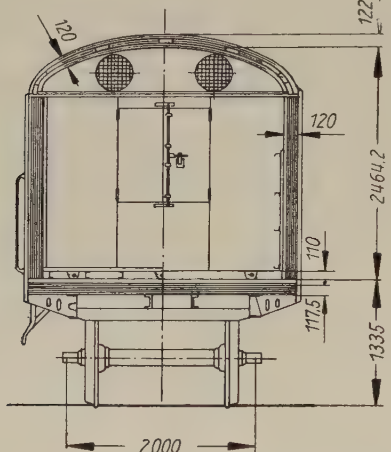


Fig. 4. — Interfrigo refrigerated wagon with two axles, U. I. C. pattern, type 1. Cross section of wagon with view of ice container partition. Scale 1 : 80.

end of the wagon. These have to be designed in such a way that they allow the water to run out of the wagon but do not allow warm air to enter from outside or cold air to escape from inside.

There is duck-boarding on the floor, latterly made of galvanised steel gratings; this is designed in such a way that there is as little resistance as possible to the circulation of air in the wagon. It is fixed to the walls by hinges. It is possible to run a forklift truck with a maximum load per wheel of 600 kg over these gratings. Owing to the thinness of the grating, dictated by the need to have as much room as possible between it and the meat hooks, it is not easy to meet the condition of high loading capacity with reduced weight of the grating.

The wagon is fitted with a single loading door on each side, 1 400 mm wide and 1 875 mm high. Single doors give rise to less points of loss than double doors, and having only one unbroken edge, are easily made air-tight. On the other hand, the suspension of the door and its closing are more difficult. The usual arrangement of ordinary hinges on the side wall is not very convenient owing to the exceptional width of the door, 1.4 m. In addition, it is not always possible to leave enough room for this 1.4 m width on the loading platform. Frequent damage to the doors may result. Experience will show how this single loading door with the new type of suspension stands up in service. It is being used here for the first time as a large scale trial on 350 wagons.

On this wagon, the trapdoors for load-

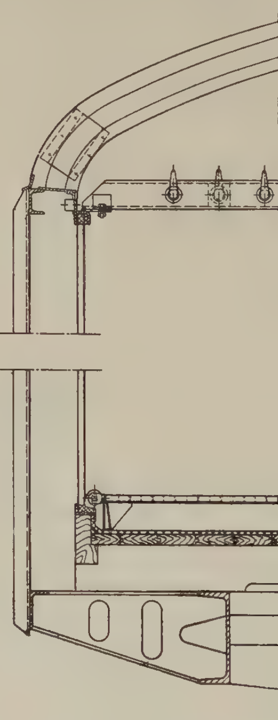


Fig. 5. — Interfrigo refrigerated wagon with two axles, U. I. C. pattern, type 1. Cross section of the side, floor and roof. Scale 1 : 40.

ing the ice are in a recess between the roof and the end wall (fig. 3). The opening of this trapdoor is so arranged (it is inclined about 30° to the horizontal) that it is possible to load the ice mechanically from above. The ice can also be thrown in by hand. For this a platform is provided with a railing on each end wall; it can be reached from each side of the wagon by ladders.

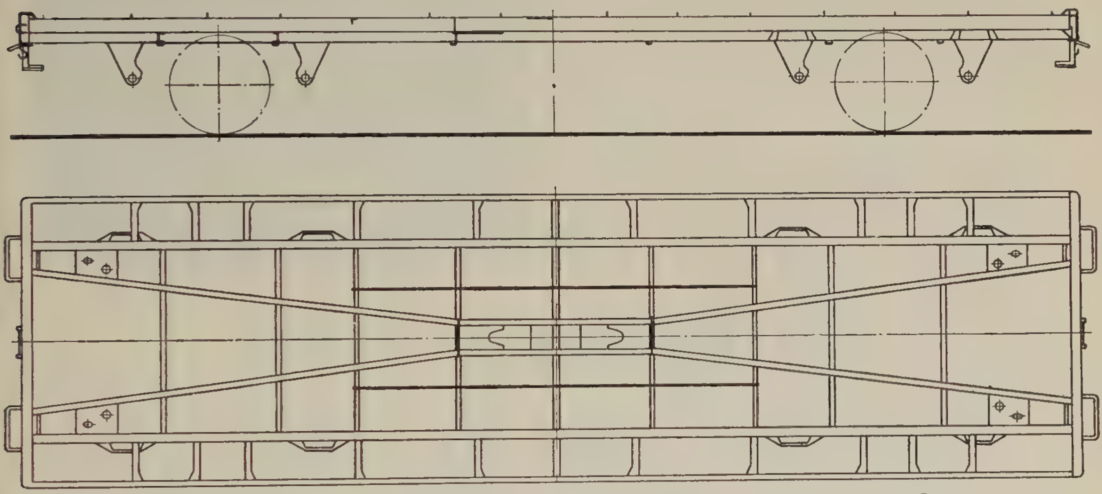


Fig. 6. — D. B. refrigerated wagon with two axles, U. I. C. pattern, type 2. Diagram of frame. Scale 1 : 80.

As on the first Interfrigo wagons, the floor is insulated by 117.5 mm thick Onazote. On the other hand, the walls and roof have been fitted with a type of insulation widely used for some time in the United States, Isoflex K 20, 120 mm thick; this consists of a mattress of glass wool 25 mm thick and sheets of corrugated aluminium stuck together and used for insulation in one, two, three or four layers. The sheets of aluminium are intended to form a single barrier layer to prevent the diffusion of water vapour. The coefficient of heat conductivity of Isoflex K 20 is 0.03 kcal/m^2 h °C., and its specific weight about 14 kg/m^3. Experience will show if the glass wool which in itself is hygroscopic will not become saturated with water in the long run in spite of these sheets of aluminium, which would considerably reduce the coefficient of conductivity. The coefficient of heat transmission of the wagons, which should be very favourable with this insulation, is not yet known as the first wagons are still under construc-

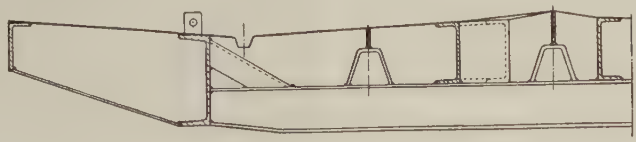


Fig. 7. — D. B. two axled refrigerated wagon, U. I. C. pattern, type 2. Cross section of frame with false floor and channel. Scale 1 : 20.

tion. However, the thickness of the insulation, 120 mm is very strictly measured.

Column 3 of the table gives the main characteristics of this refrigerated wagon.

D. B. and Transthermos, model U. I. C., type 2, refrigerated wagon.

For the transport of refrigerated goods, Interfrigo uses the Transthermos universal refrigerated wagons. These all-metal wagons when new have a coefficient of heat transmission equal to not more than 0.3 kcal/m^2 h °C., and are suitable for the transport of all refrigerated goods, even those which have to be carried at very low temperatures. They are the type 2 U. I. C. wagon, several hundreds of which are already in service on the D. B.

The frame, which as we have already said, is used for both types 1 and 2 of the U. I. C. wagon, is made up of welded rolled sections (fig. 6). Only sections on the U. I. C. list have been used. The wagons are subjected to the international compression test of 100 t applied to the buffers and 40 t diagonally, also applied at the buffers, as laid down. The stresses set up during these tests did not exceed the admissible values. They were relatively low. In addition, on the wagons of this type which have already been in service for four years, no defects in strength nor any damage due to this cause have been noted in either the frames or bodies of the wagons.

The false floor of 1.25 mm sheet is welded to the frame. This steel sheet to which 0.2 to 0.3 % copper has been added resists corrosion better than ordinary steel. The false floor carries the floor insulation. It is inclined in the transversal direction of the vehicle towards a channel the whole length of each of the side sills (fig. 7). The channels slope towards a water drain.

The body of the wagon is laid on the frame covered by the false floor. The side walls and the end walls consist of 1.5 mm thick steel sheet welded to the body supports of vertical and horizontal angles (fig. 8). Owing to the design adopted, the side wall has a great resistance to flexion and helps in carrying the load.

The roof, which is quite separate from the body (fig. 9), consists of steel sheet 1.25 mm thick and angles. All the parts are welded together. In the end wall, near the roof, is the trapdoor for loading the ice, and in the middle of the roof, four openings for the Flettner rotor fans. The outer sheets of the body, the roof and the false floor are hermetically welded so as to effectively prevent the diffusion of water

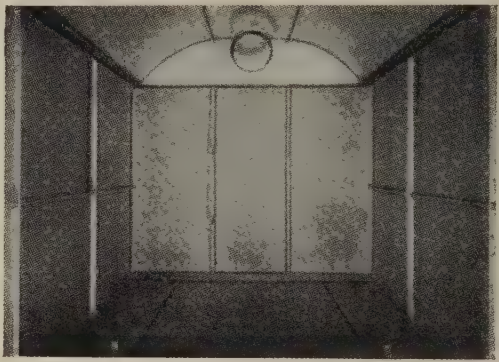


Fig. 8. — Inside view of outer covering of D. B. refrigerated wagon.

Fig. 9. — Inside view of roof of D. B. refrigerated wagon without the insulation and interior covering.

vapour which might reduce the coefficient of heat transmission of the insulation.

When the main construction of the body is completed, the inside painting is done.

Fig. 10. — Inside view of floor of D. B. refrigerated wagon with cross members and floor frame, without insulation.

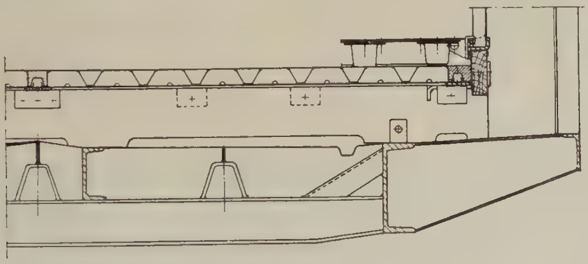


Fig. 11. — Steel frame and floor of D. B. refrigerated wagon in section. Scale 1 : 20.

To prevent rust forming, three coats of bitumen 2 000 μ thick in all, are applied. The oak cross members are solidly fixed to the solebars at the ends by bolts and flat iron joint covers (fig. 10). The floor frame, which goes right round the floor, is fixed on the beams and the interior walls rest on it. The floor consists of six hollow galvanised sheet caissons which are bolted onto the cross bearers; the joints between the caissons are filled with bitumen. This floor (figs. 11 and 12) is capable of supporting, as recently laid down by the U. I. C., the maximum load per wheel of 12 t when fork lift trucks are used on it. The floor is hollowed out across the ends to form a trough to collect the water from the melting ice, lined with light metal (fig. 13) to which two drain pipes are welded to carry off the water outside the wagon.

The inside walls are formed of corrugated aluminium sheet 2 mm thick, rivetted at top and bottom to *h* section aluminium profiles press formed. The stiffness of the wall due to the corrugations is so great that it is sufficient to rest it against the outer stressed steel body at the top against the roof and at the bottom, near the floor (fig. 14). This arrangement avoids points of loss. In order to be able to remove occasionally the inside linings in order to check the insulation, the lower *h* section is screwed to the floor framing so solidly that it is easy to remove the wall without taking up the floor. The upper *h* section is screwed

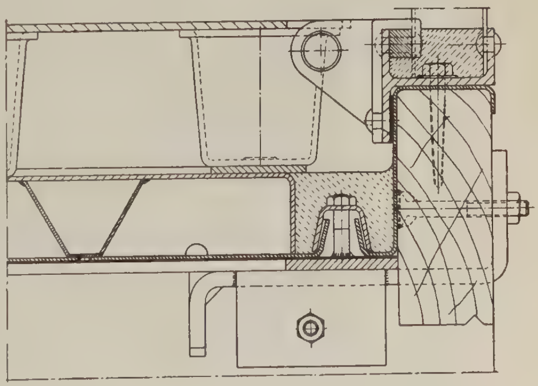


Fig. 12. — Metal frame and floor in section; part of figure 11 enlarged 5 times.

to the extended ends of the inner steel roof sticks (fig. 15). The inner roof sticks are fixed to the outer steel one through wooden packing blocks. The roof is covered on the inside with smooth aluminium sheet (1.5 mm thick). The sheet is crewed to wooden packings carried on the inner roof sticks (fig. 16).

The 16 meat bars of the usual figure 8 shape each carries 16 hooks, the bars being

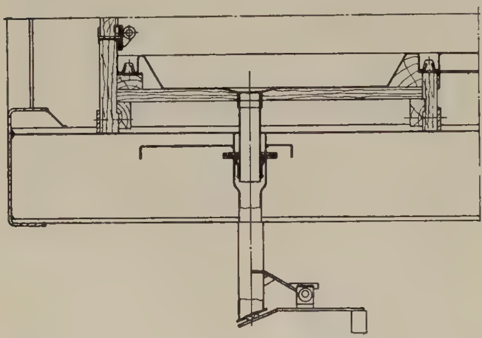


Fig. 13. — Trough for ice water of D. B. refrigerated wagon (longitudinal section of wagon). Scale approx. 1 : 20.

supported by the upper *h* section (fig. 17). The hooks are of forged aluminium alloy. The arrangement of the meat bars across the wagon transversely, adopted for more than 30 years on the D. B., has proved extremely satisfactory. It makes it easy to support the bars. When the meat hooks are arranged longitudinally, the hooks and the carcasses hanging from them are likely to move along the longitudinal axis of the wagon if it is shunted carelessly. Using transversal bars it is possible to slide the hooks along and bring them up close to the body sides, so that the whole loading space is fully utilised when other goods are also loaded. According to the latest regulations, U. I. C. pattern refrigerated wagons types 1 and 2 must have at least 300 meat hooks.

From the general point of view, the use of light alloys was absolutely essential in such proportions for the tare weight of the heavily insulated wagon not to exceed 16 t. The interior lining of light metal

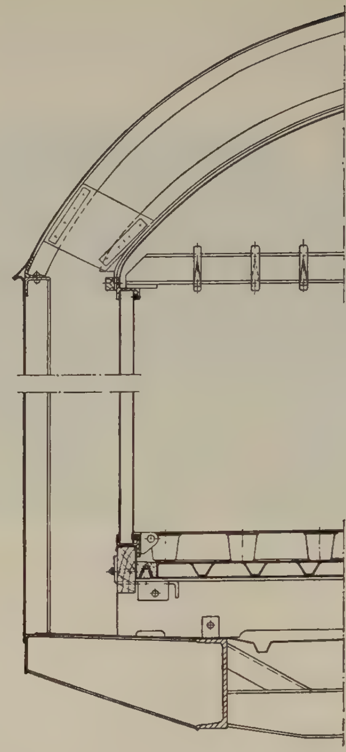


Fig. 14. — Section of cross partition of D. B. refrigerated wagon. Scale 1 : 40.

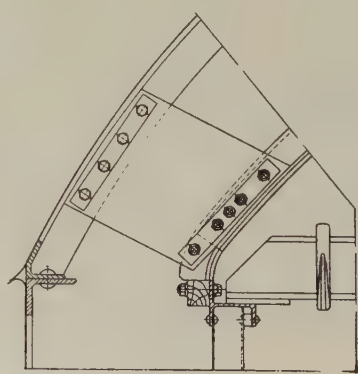


Fig. 15. — Upper part of side wall of D. B. refrigerated wagon with connection to roof. Part of figure 14 enlarged 2 times.

for the walls and roof has the further advantage of giving, a hygienically correct and clean interior surface. The anodising of the aluminium parts prevents any corrosion, and the clean appearance of the inside surfaces is permanently retained. The persistence of the smell left by certain goods or damaged consignments, which is often a nuisance with wood-lined refrigerated wagons, is thus done away with. Moreover, no painting of the surfaces of the loading compartment is necessary.

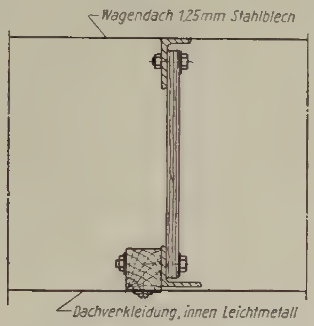


Fig. 16. — Longitudinal section through roof stick of D. B. refrigerated wagon.

N. B. — Wagendach... = wagon roof, 1.25 mm sheet steel. — Dachverkleidung... = interior roof covering, light metal.

According to the U. I. C. note, a refrigerated wagon must have an ice container along the whole length of each end, of a capacity of 1 250 kg. The ice container, based on these data, with a volume of 2.53 m³ is formed by the end of the wagon, its sides and the movable front partition. In the part occupied by the ice container, the ends and sides are covered with aluminium sheet. As a result the ice cannot come directly in contact with the partitions, thereby impeding the flow of air. The hot-galvanised steel grills which support the ice boxes are fitted about 50 cm above floor level.

The movable front partition of the ice container has to be very robustly made. It is formed of particularly strong light metal sections and corrugated light metal sheet with deep corrugations. It has two

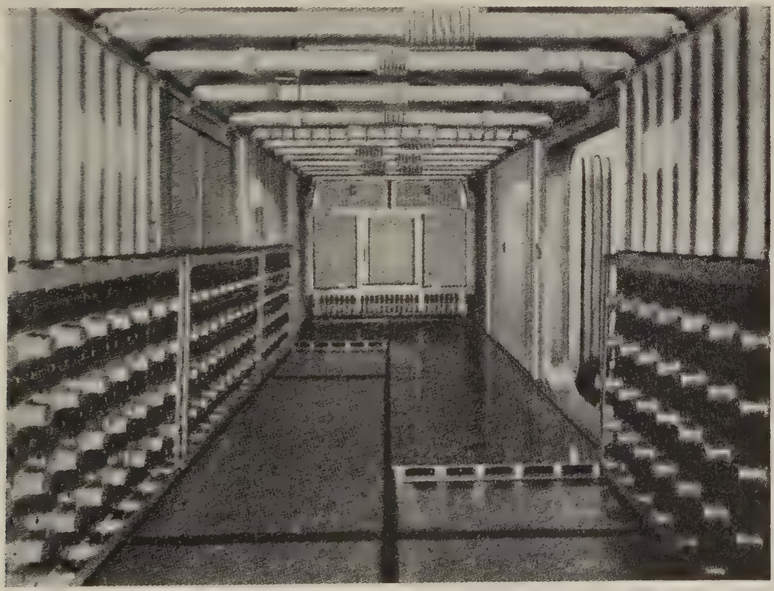


Fig. 17. — Interior view of D. B. refrigerated wagon with duck-boarding partly lifted up against wall.

shutters in the upper part and a door in the middle of the lower part (fig. 18). When dry ice is being used, or when it is merely being used as an insulated wagon, the two front partitions of the ice containers are pushed back against the ends. This gives an additional space of 1.35 m² at each end, or a total extra 2.7 m² loading space, when heavily frozen goods or goods merely needing thermal protection are being carried. The clients of the D. B. attach the greatest importance to this increased loading space.

At the top of the roof, arranged in the longitudinal direction of the wagon, is the dry ice tank made of aluminium sheet which is necessary for transport at very low temperatures. This tank is divided into two parts in the middle of the wagon by the Flettner rotor fans. Each half can take 425 kg of dry ice which is loaded from the end. The regulator provided at the end of the duct (fig. 19) is connected to the ice container partition. When this partition is moved forward, the regulator automatically uncovers an opening through which the ice can fall into the container. At the same time, a grill (fig. 20) automatically closes the entrance to the dry ice tank. When dry ice is being loaded, the ice container partitions are pushed back against the end of the wagon, so that there is in fact no ice container.

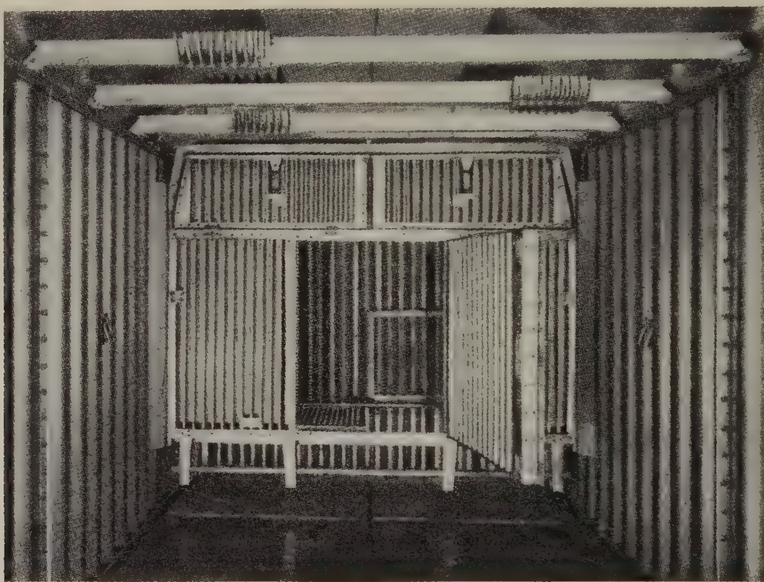


Fig. 18. — Ice container of D. B. refrigerated wagon, door open.

The natural circulation of air in the wagon is helped by four Flettner rotor fans (fig. 21). The fans draw in the air in the middle of the wagon and send it through the dry ice tank duct towards the ice containers (fig. 22). Here the air is cooled, falls to the lower part and returns through the duck-boards on the floor to the middle of the wagon. The output of these four fans has so far been found sufficient in service on the D. B. A wheel in the middle of the rotors makes it possible to reduce the circulation of air when dry ice is being used, or even to shut it off completely. This is often desirable or necessary because too fast a consumption of the dry ice would lead to unduly low

temperatures. For this purpose, two double shutters have been provided in the central part which separates the space occupied by the fans of the two longitudinal ducts to take the dry ice. At each end of the

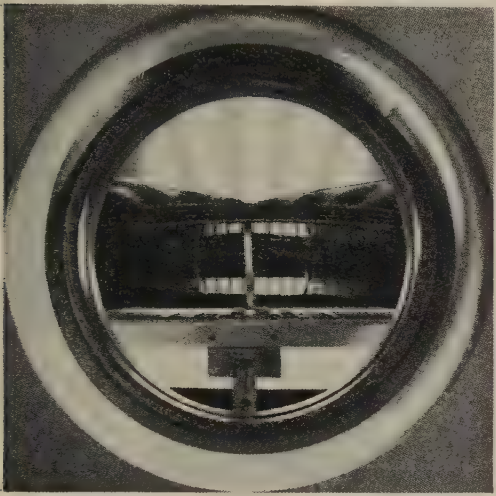


Fig. 19. — View of dry ice duct of D. B. refrigerated wagon.

wagon in the sheet at the end of the longitudinal duct there is a regulator with slots (fig. 19) which allows the cold current from the dry ice to flow into the loading compartment. Thanks to double shutters, it is possible to regulate the amount of cold given off by the dry ice. In conjunction with the regulators, it is possible in these universal refrigerated wagons to regulate the circulation of air as required by three positions of working :

1° refrigeration by natural ice : double shutter in the position « ice refrigeration » (fig. 23*a*);

2° limited refrigeration by dry ice : double shutter in the position « dry ice refrigeration — normal » (fig. 23*b*);

3° intensive dry ice refrigeration : double shutter in the position « dry ice refrigeration — strong » (fig. 23*c*).

Attention is drawn to these three positions in the wagon by a notice in three languages : German, French and Italian, adjacent to the lever for operating the double shutters.

When the wagon is used as an insulated wagon, the double shutters must be placed in the second position, because in this position the effect of the fans is annulled.

The duck-boarding is fixed to the inside side walls and can be removed (fig. 17 and 24). It has been calculated for a load per wheel of the fork lift truck of 1 200 kg and weighs 37.5 kg. To obtain good air circulation, the space under the duck-boarding must be as great as possible. Its height of 80 mm is limited by the space needed between the top of the duck-boarding and the bottom of the meat hooks to give room to move about in the wagon.

In the middle of the wagon, on each side, is a loading door with a clear opening of 1 500 × 1 800 mm mounted in a frame welded to the side wall to increase rigidity. The doors are double doors and are made air-tight by means of a fourfold rubber joint (fig. 25). To conserve air-tightness, the corners are rounded.

On the German Federal Railways, the air-tightness of the body is checked in the

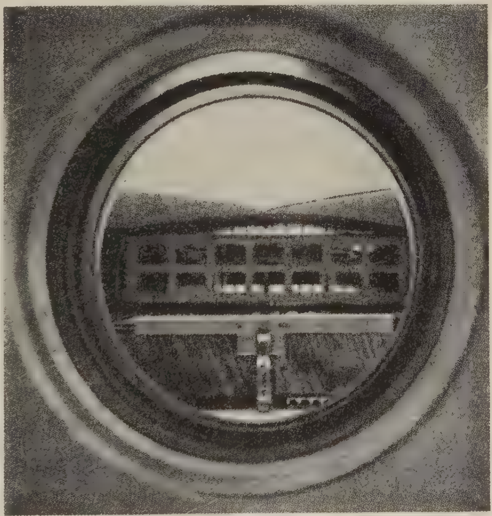


Fig. 20. — View of dry ice duct with hydrous icing.

following way: air is forced into the wagon through one of the devices for draining off the ice water and an excess pressure of 6 mm water compared with the outside

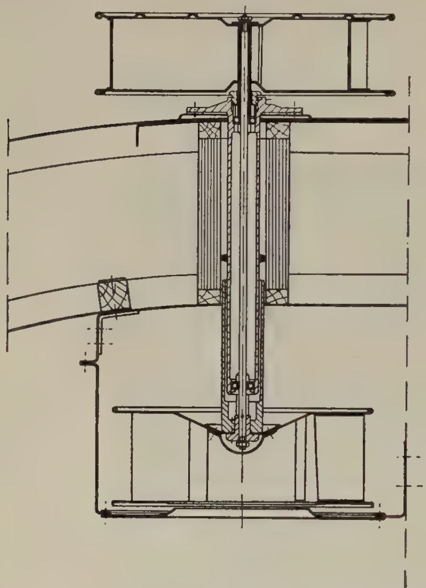


Fig. 21. — Diagram of Flettner rotor fan.

pressure maintained. Under these conditions, loss of air through defects in airtightness of the universal refrigerated wagons are only 19.8 m³/h, or 0.34 times the volume of the wagon, in other words it is extremely good.

In each end wall, there is a trapdoor for loading the ice. For the sake of airtightness, this has been made cylindrical. It can be reached from a service platform (fig. 26). When the ice is loaded mechanically, a spout with an elbow joint has to be used to guide the ice into the container.

At each end of the wagon are two devices for draining off the ice water, and as a precaution two devices for draining off any condensation water from the insulation (fig. 27). Although the body of the wagon is hermetically welded, it is not possible to prevent condensation water forming in the long run. It might be possible in the future by making some sort of drain in the insulation space. This depends on the results given by trials now in hand on the dehydration of the insulation by the Minikay process. With this process, which is already used on ships and fixed refrigeration installations, a circulation of air is set up in the insulation, either forced by means of fans, or naturally by the difference in temperature. The humidity contained in the circulating air is constantly precipitated over a cooled surface — the « dehydrator » — and thus drained off from the insulation space.

The insulation is 250 mm thick in the sides and roof, and 160 mm in the floor. Figure 28 shows how it is constituted. The insulating materials used are Isoflex, which has already been used for the roof of the first 525 Interfrigo wagons, and Iporka-

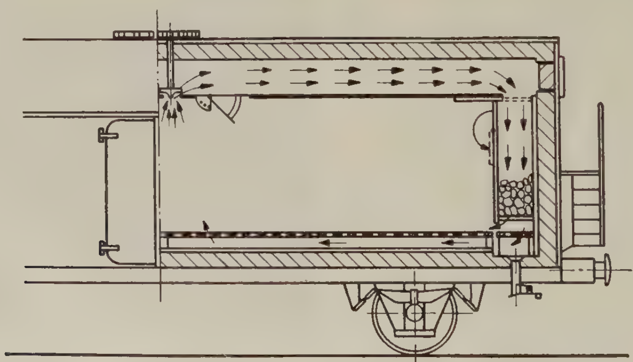


Fig. 22. — Diagram of circulation of air through wagon.

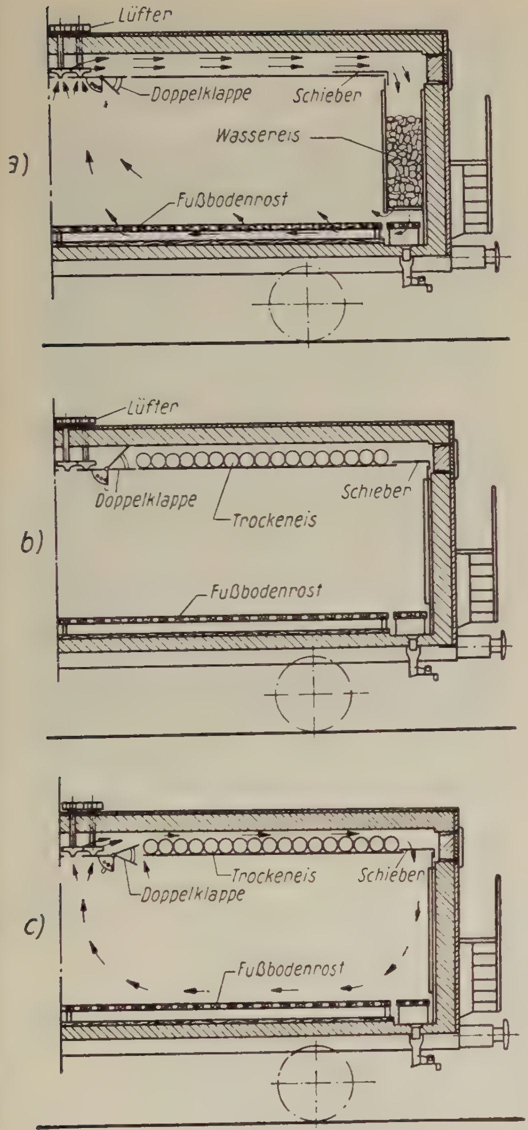


Fig. 23. — Service position of double shutters for regulating the circulation of air : *a*) position : « ordinary ice refrigeration »; *b*) position : « dry ice refrigeration, normal »; *c*) position : « dry ice refrigeration, heavy ».

N. B. — Lüfter = fan. — Doppelklappe = double shutter. — Schieber = regulator. — Wasser = ordinary ice. — Trockeneis = dry ice. — Fussbodenrost = duck-boarding.

F. L. G. The latter product consists of flakes of Iporka tightly pressed to give a volume weight of 30 kg/m³ enveloped in a sheet of plastic material which is welded and completely air-tight. Iporka is an artificial foam with an urea-formaldehyde base. The coefficient of heat conductivity of Iporka-F. L. G. is 0.027 kcal/m² h °C.

When the heavily insulated wagon is used for ordinary refrigerated goods, thanks to this insulation it is unnecessary to stock up again with ice during long runs. The transport time is thereby improved, which saves both time and money.

The main characteristics of the D. B. refrigerated wagon are shown in column 4 of the table.

Other types of wagons and refrigeration equipment.

a) *Special experimental type for the Transthermos Company.*

At the request of the Transthermos Co., and based on suggestions made by it, the D. B. designed and built in 1952 two all-steel refrigerated wagons which differ from the type of refrigerated wagon generally adopted. On these latter, in effect, it is the outer covering of the body which is stress-carrying and takes the insulation and inner covering. On the wagons designed according to the Transthermos suggestions, on the contrary, the inner sheeting is the stress-carrying component on which the insulation is fitted and the outer sheeting is merely a covering. The method of construction suggested required a considerably longer time to make, and in the opinion of the shops, even if mass-produced, would take longer to make than the standard type of refrigerated wagons. It was also a departure from the usual type of insulation used on the D. B. which is well proved, in the sense that the floor is only insulated with Iporka-F. L. F. which has the good coefficient of heat conductivity of 0.07 kcal/m² h °C. Experience has shown that it is better to avoid using this material in the insulation of the D. B. refrigerated wagons because humidity is particularly bad

for it. The D. B. therefore in this case only used Isoflex whose coefficient of conductivity is 0.04 kcal/m² h °C. With Iporka-F. L. F., it is true it has been possible to obtain for the two wagons a coefficient of heat conductivity of K = 0.28 kcal/m² h °C. However, as was to be expected, this value of K has already increased after a year and a half to 0.35 kcal/m² h °C. When measuring periodically the coefficient of heat transmission in these wagons, which will shortly be done, it is practically certain that the value of K will be found to have increased again. During compression the values of K for the two wagons were measured once more: they amounted to 0.46 and 0.55 kcal/m² h °C. The increase of 7 % in loading capacity compared with the universal refrigerated wagon is obtained solely by reducing the thickness of the insulation to 200 mm. The results obtained with these wagons have shown that it would not be possible to fulfil with them the conditions laid down by the U. I. C. for the type 2 wagons.

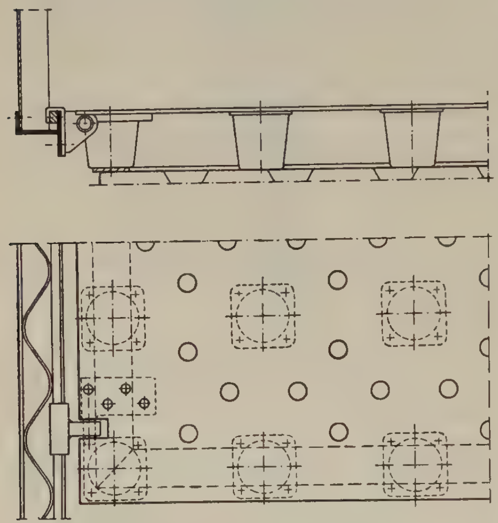


Fig. 24. — Diagram of removable light metal duck-boarding of D. B. refrigerated wagon. Scale about 1 : 25.

b) *Insulated wagons for carrying bananas.*

For carrying bananas, a traffic which has been growing considerably for some time, it is not necessary to use refrigerated wagons of the types described above. Lightly insulated wagons where the temperature remains constant are all that is needed. On the German Federal Railways, the old refrigerated wagons have been mainly used for this purpose, but there is no longer a sufficient number of them. At the request of the Transthermos Company, an ordinary covered wagon has been used which has been insulated by covering the walls with a 100 mm thick layer of stryropor, a well-known product, though unfortunately only available at present in an extremely in-

Fig. 25. — D. B. refrigerated wagon, loading door open.

flammable form. It is proposed to insulate 100 covered wagons in this way to start with. It remains to check with care if it is not possible to increase the loading pos-

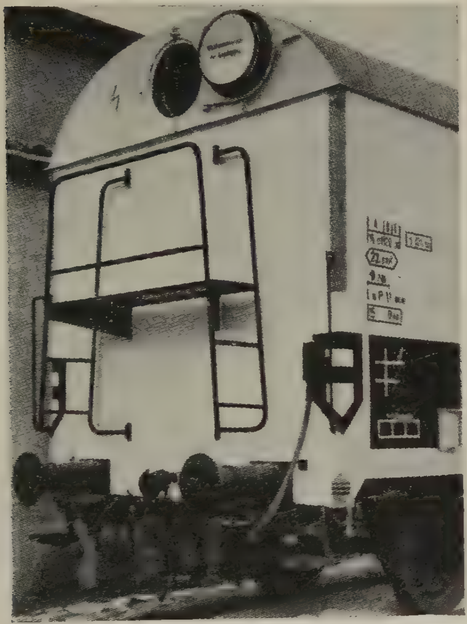


Fig. 26. — End view of D. B. refrigerated wagon, trapdoor for loading ice open.

sibilities by fitting an intermediate floor, allowing of loading in two tiers. A bunch of bananas weighs up to 60 kg, and may be about 1.2 m high. After fitting the

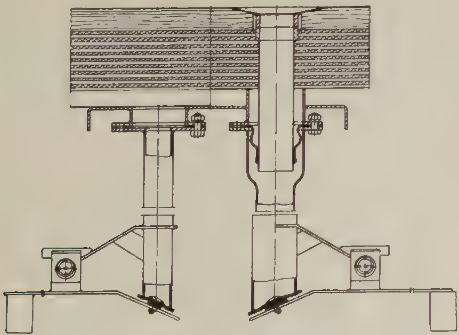


Fig. 27. — Diagram of arrangement for carrying off water in D. B. refrigerated wagon.

insulation and intermediate floor, there is a free space of about 2.4 m in the covered wagon to the roof at the centre and 1.8 m at the eaves. It would therefore not be possible to load bunches of bananas vertically as the clients wish.

c) *Pre-refrigeration by means of movable fans.*

Whilst Interfrigo has experimentally equipped some of the 350 wagons now under construction with electric fans which

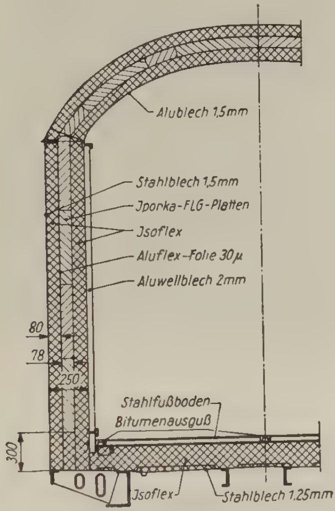


Fig. 28. — Constitution of insulation of D. B. refrigerated wagon. Scale approx.: 1 : 50.

N. B. — Alubech = aluminium sheet. — Stahlbech = steel sheet. — Aluwellblech = corrugated aluminium sheet. — Stahlfussboden = steel floor. — Bitumenausguss = bitumen layer.

can be used not only during the run but also and above all for the pre-refrigeration of the wagons, the Transthermos Company has designed for this purpose a relatively simple and economic method. In this case two fans are suspended through the loading trapdoors above each ice container, having a power of approximately 500 W each. These fans reinforce the natural circulation of air. They each have an output of 4 600 m^3/h air when working freely. The fans, shown in figure 29

near a wagon, are introduced into the wagon through the ice-loading trapdoor (fig. 30) and suspended above the ice container (fig. 31). The current is supplied

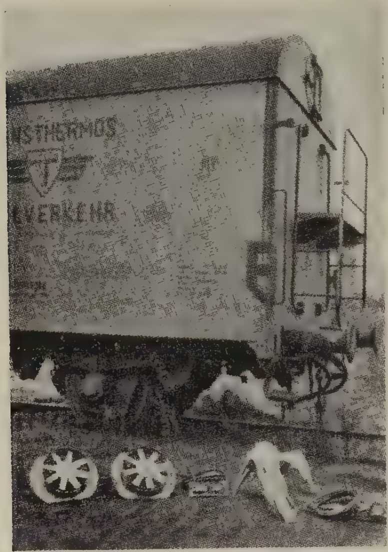


Fig. 29. — Set of fans before installation.

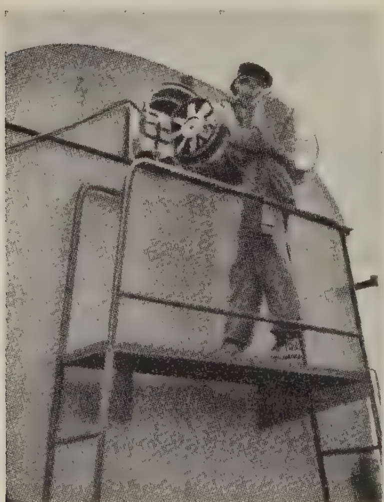


Fig. 30. — Fitting the fans.

from a cable passing through the trapdoor which is left slightly ajar.

Refrigerated containers.

In addition to refrigerated wagons, refrigerated containers are now used for carrying perishable foodstuffs and other refrigerated goods. The container traffic, which has grown more and more in recent years, is a special element in the refrigerated transport chain, because it allows of the transport of goods from door to door, even when the firm has no private siding connecting it to the railway. These containers, which can be loaded or unloaded at the cold store expose the goods to only a minimum fluctuation in temperature. A

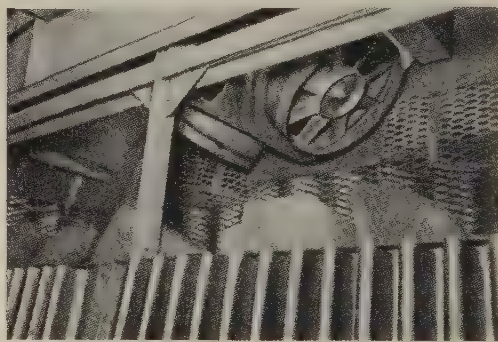


Fig. 31. — Fans in position above ice container.

high technical standard has been reached in the construction of these refrigerated containers, so that they have a low weight and a good coefficient of heat transmission. It is easier to respect the laws governing the technique of refrigeration in their construction than in the case of wagons, whilst keeping down the weight, owing to the small superficial area of the walls; it is easy to make the walls play their part in the strength of the whole without having to have a stressed framework, and consequently without increasing the weight.

The new « pa » [2] refrigerated con-

[2] The shortened designation « pa » (= wagon built to carry containers) characterizes the method of transport for containers.

tainer of the German Federal Railways (fig. 32 and 33) is so designed that it can be transported by the carrying wagon (B. T. wagon) on the railway and on the

Fig. 32. — « Pa » refrigerated container of the D. B.

road vehicles of the D. B. It is also possible to transport and load it on a railway or road vehicle of suitable size with suitable lifting gear. It can be moved or

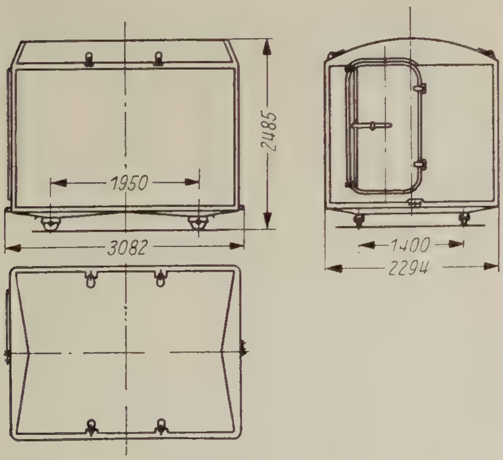


Fig. 33. — Leading dimensions of the « Pa » refrigerated container of the D. B. Scale 1 : 100.

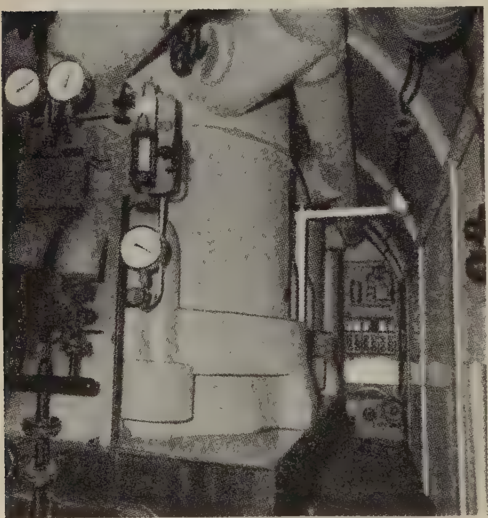
loaded by means of a crane and meets the U. I. C. regulations concerning customs regulations and the regulations governing international refrigerated transport of perishable foodstuffs.

The « pa » refrigerated container (series Ei mark) is designed for all types of goods affected by heat and lightly packed. It rests on its own running gear which consists of a flat frame made of longitudinals and crosspieces to which four wheel guards with their rollers, four eyes and two couplings are fastened. The outer covering of the container is welded to the frame. The rollers are 1 400 mm apart and their wheelbase is 1 950 mm. The diameter of the rollers is 200 mm and they are 75 mm wide.

The body of the container consists of an exterior case, and an inner box with insulation between the two. The body of the container is sufficiently strong to stand the effects of lateral shocks, as laid down in the U. I. C. regulations governing the strength of containers.

The outer case is of 1.25 mm sheet steel, gas-welded. It is reinforced internally by inverted U sections surrounding the sides and roof and also supporting the four rings by which the container is lifted by crane.

The inner case is made of completely air-tight gas-welded aluminium sheet 1.5 mm thick for the walls and roof, and 2.5 mm thick for the floor. The two sides and the end opposite the door are lightly reinforced by angles. All the walls are stiffened by means of inverted U sections, point welded, which in addition, in combination with the floor duck-boarding, assure good air circulation. Special containers have been provided to take dry ice. The inner case has no fixed connection with the outer case of the container. It is embedded in the insulation without any points of loss. The inner case is 2 560 mm long, 1 880 mm wide, and 1 850 mm high above the duck-boarding. When carrying delicate goods which must not be crushed, intermediate floors and partitions can be fitted in the container.



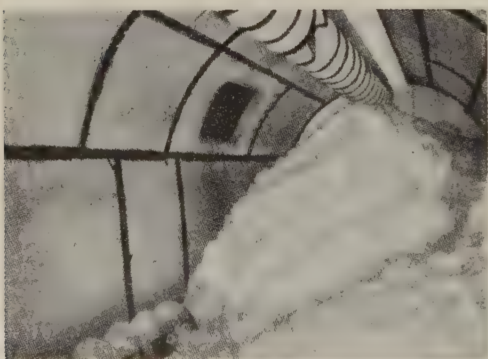


Fig. 34. — Mobile ice-making plant.

The loading door is a single door 1 655 mm high and 850 mm wide, made air tight by means of rubber sections and closed by means of a turn screw.

The insulation is 160 mm thick. The walls and roof are insulated by means of Iporka-F. L. G. sheets in Igelith casings, the floor and door by Polystyrol foam. The K value of the container is about 0.2 kcal/m² h °C.

Ice-making installations.

If at the pre-refrigeration points, which according to the locality and nature of the client's business, may lie in very scattered places, there is difficulty in obtaining supplies of ice, or if the local price of ice is too high, it is possible to remedy this by using mobile ice-making equipment. For this purpose, the Transthermos Co. has installed on a low-loading four axled wagon a very economic ice-making installation, capable of producing 25 t of ice a day (fig. 34).

For making pulverised ice, which is used above all when transporting fresh vegetables, use is made of machines which pulverise the ice together with a projection fan (fig. 35). The ice is milled from blocks and blown over the load. To complete the direct icing, the ice containers are loaded with slabs of ice. With this method of refrigeration, the vegetables remain fresh and look well even after long journeys.

Future perspectives of developments.

Refrigerated transport has made considerable progress during recent years. In view of the increase in the population from year to year and the economic growth which it is to be hoped will be durable, still greater development of this method of transport is to be expected. Such expectations are not unjustified when it is remembered that in the United States the railways have some 135 000 refrigerated wagons for their refrigerated transport. This figure seems particularly high when it is compared with the 18 000 refrigerated wagons

now existing in Europe where the population is so much greater. But the difference does not lie in numbers alone; it must also be remembered that the European refrigerated wagons are usually four wheelers, whilst those of the United States are bogie wagons and consequently have a much greater load capacity.

To date we have succeeded in meeting the ever-growing traffic with the refrigerated wagons described above. The capacity of the new existing refrigerated wagons could be increased if it were possible to make up complete trains of refrigerated wagons running at 100 km (62 miles)/h. If complete trains could be made up, it would also be economically defensible to use mechanically refrigerated wagons, in which case the carefully calculated refrigerating machinery of the different wagons could also be used for the pre-refrigeration of the load, thus making possible a further reduction in the time of transport from producer to consumer. Wagons with mechanical refrigeration have the great advantage of being very quickly pre-refrigerated, even whilst being taken to the loading point; in addition this does away with restocking with ice during the run, which may save a lot of time compared with the use of the usual refrigerated wagons. In addition, it is much easier to regulate the temperature in wagons with mechanical refrigeration than in those using ordinary or dry ice, an advantage which must result in lowering the transport costs seeing that on the one hand there is less risk of damage to the load and on the other no excessive consumption of refrigerating products. The construction of a wagon equipped with a refrigerator raises very interesting technical problems, especially as regards the regulation of the temperature. Such wagons have already been in use for some years, it is true, but in a very sporadic way. Moreover an employee has to travel in them, which has a very bad effect on the cost of the transport. This problem could be solved more easily if complete trains of refrigerator wagons could be formed, because in that case one specially trained employee would suffice to supervise the condition and temperature of the goods throughout the wagons of the train. About three years ago, the question of using wagons with refrigerators was gone into once more. A new all-metal refrigerator wagon was built by the Waggonfabrik Rathgeber in conjunction with the firm of Brown Boveri & Co. (Cold Section); it remains to be seen, however, how this will stand up in service from the technical and economic points of view.

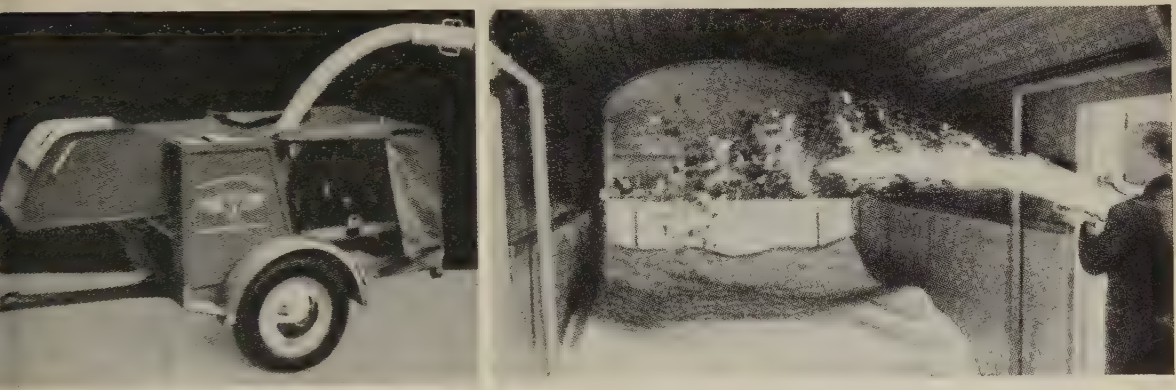


Fig. 35. — Machine for pulverising and projecting the ice.

If the two refrigerated wagons, types 1

and 2 described above, are compared with those built during the last twenty years, which unfortunately are still largely in service, it must be admitted that enormous progress has been made, especially in the last few years, in the field of refrigerated wagons both in Germany and in other countries. By introducing new regulations and stricter clauses the U. I. C. and O. R. E. have made sure that all the most recent progress in the construction of rolling stock and in refrigeration technique will be applied in future in the construction of new refrigerated wagons. For example, the running gear has been improved by the adoption of long suspension springs and roller bearings to such an extent that the modern refrigerated wagons can run at a speed of 120 km (74 miles)/h. With regard to the compressed air brake, the use of a type allowing a running speed of 100 km (62 miles)/h has been decided; it is recommended to install the automatic braking of the load in order to be able to include these wagons in trains running at a speed of 120 km/h. The frame and body must be very strong, perfectly air-tight, and also as light as possible, so as not to exceed a tare of 16.5 t. The weight of the load must correspond to the maximum laid down for the B lines with an axle load of 18 t. Such important dimensions for technical and commercial operation as the wheelbase, the length of the wagon, the usable floor area, the height of the doors, their width and their position above rail level, are laid down in a standard fashion. Directives are given about the insulation and air circulation, and the coefficient of heat transmission of the wagon is fixed. The metal duck-boarding must be jointed to the body; it must be strong enough to stand a load of at least 1.2 t. The number, arrangement and capacity of the ice containers, as well as the position of the ice loading trapdoors are all laid down.

O. R. E. goes still further and in addition to the regulations which must now be adhered to in designing the U. I. C. types 1 and 2, lays down in a precise fashion the constructional details of nearly all the components of refrigerated wagons of these two types. Particular importance is attached to perfect air-tightness, uniformity of temperature throughout the loading compartment, proper air circulation, low thermal inertia, rapid and thorough cleaning, and the absence of any lasting smell from the goods carried.

It is thus certain that all the refrigerated wagons put into service in Europe in the future will include all the most recent improvements made from the point of view of vehicle construction as well as of refrigeration technique. Theis standardised construction makes it certain that spare parts will be available throughout Europe should any parts be damaged. This will prevent wasteful time out of service. The refrigerated wagons will thus become a powerful link in the refrigeration chain from producer to consumer. It is only to be hoped that the railways of Europe will soon be in a position to go over their old refrigerated wagons which are unsuitable but unfortunately still very numerous, and replace them by the new types. This is the only way ot achieve really satisfactory refrigerated transport by rail which will stand up to competition.

[625 .234]

Heating arrangements with Diesel traction.

Economic consideration and a comparison of various methods of heating when using Diesel traction,

by Siegfried BÜTTNER,

Institution of Railway Rolling Stock, Berlin-Adlershof.

(*Deutsche Eisenbahntechnik*, No. 2, February and No. 4, April 1957.)

I. Introduction.

1.1. *General.*

One of the most difficult problems which arises with the introduction of Diesel traction is that of heating. For more than a century, railway traction has been mainly performed by steam locomotives so that the heating of passenger trains has not presented any special difficulty. The steam needed for heating is taken from the locomotive by means of a steam valve and distributed through heater pipes. Because of the simplicity and economy of this method of heating, the great majority of coaches in service are still fitted with steam heating equipment.

The electrification of various railway lines began in 1910, led to the introduction of heating by electric resistance, so that there are at present passenger carriages in service with both types of heating [1]. Recently, the German Federal Railways have put into service experimentally some carriages with hot water heaters, in which the heating liquid is brought to the required temperature in heat exchangers fed by the normal steam or electric heating [2]. The most modern railway carriages, particularly multiple-unit sets, are often equipped, not with ordinary steam or electric heating, but with fully automatic air conditioning : this applies, for example, to the lightweight motor coach articulated set of the Deutsche Bundesbahn [3].

Another fairly widespread type of heating is that using hot water from a special boiler on each vehicle. Whilst this, in Germany, is used only on rail ars, almost all long distance carriages of the Soviet Union, China and Korea are fitted with it [4].

Excepting the types of heating with boilers in the carriages, all other types have a central boiler or energy producer. This central equipment can be placed in a separate vehicle, inserted in a train as required, or can be installed on the Diesel locomotive. In the latter case, the limitations in regard to weight and space demand that the boiler or heating current producer should be as compact as possible, which means the compliance with conditions which are not met with in the case of similar fixed equipment.

Apart from a construction which must be as compact and as light as possible, heating equipment must provide fully automatic control, reduced maintenance, rapid starting and easy inspection.

1.2 *Present conditions and purpose of the investigation.*

The types of heating used today have all reached a high degree of technical perfection, so that in the selection of a type to be used the level of technical development no longer has a prime importance, this place is occupied by economic considerations. Because it is possible to provide on the Diesel locomotive, in a very small space, a boiler or generator working entirely automatically, it cannot be claimed that the steam locomotive retains an advantage in regard to heating.

To show the advantages and disadvantages of various train heating equipments, a comparison will be made in this article of the following systems :

— electric heating of the train with heating current supply equipment installed on the Diesel locomotives;

— steamheating by Diesel-oil fired boiler mounted on the Diesel locomotive;

— steamheating by oil fired and exhaust gas boiler mounted on the Diesel locomotive;

— hot water heating with electric heat exchanger and equipment to supply the heating current on the Diesel locomotive;

— hot water heating with steam heat exchanger and oil-fired boiler on the Diesel locomotive;

— hot water heating with steam heat exchanger, exhaust gas and oil-fired boiler on the Diesel locomotive;

— hot water heating with coal fired boiler for each carriage;

— hot water heating with oil-fired boiler for each carriage.

The comparison will allow the selection of the most economic method of heating and the most suitable for the Administration concerned.

2. Calorific requirements for heating passenger carriages.

By analogy with the determination of calorific requirements of buildings according to the principles of standard DIN 4701, we may describe the calorific needs of a carriage as :

$$Q_i = Q_o \, (1 + Z_H + Z_W + Z_G + Z_L)$$

$$\frac{t_i - t_{am}}{t_i - t_{a\,max}} \cdot Z \text{ [kcal]} \quad . \; . \; . \quad (1)$$

in which :

Q_o = sum of the heat dissipation when standing without change of air in kcal/h,

Z_H = addition for the position,

Z_W = addition for action of wind,

Z_G = addition for speed of train,

Z_L = addition for heat dissipation due to air change,

t_i = desired interior temperature in °C,

t_{am} = average exterior temperature during the period of heating in °C,

$t_{a\,max}$ = maximum external temperature and

Z = No. of hours of heating.

Q_i signifies the minimum, and therefore ideal, amount of heating needed to reach and maintain a given temperature t_i inside the carriage.

2.1. *Loss of heat Q_o when standing.*

The losses of heat which occur when standing without change of air are obtained from the total area of the vehicle F in m² and the co-efficient of heat transmission k_R in Kcal/m² h °C, by means of the equation :

$$Q_o = k_R \, F \, (t_i - t_{am}) \text{ [kcal/h]} \quad . \; . \; . \quad (2)$$

The resultant co-efficient of heat transmission k_R is composed of :

$$k_R = \frac{k_{Bod} \cdot F_{Bod} + k_{stwd} \cdot F_{stwd} + k_D \cdot F_D + k_F \cdot F_F}{F_{Bod} + F_{stwd} + F_D + F_F} \qquad (3)$$

According to WINOKUROW (5, p. 600), we can use for modern all-steel thermally insulated carriages the following co-efficients of heat transmission :

floor	0.9	kcal/m² h °C
roof	0.95	kcal/m² h °C
side walls	1	kcal/m² h °C
windows	3	kcal/m² h °C

These values give a resultant co-efficient of about 1 kcal/m² h °C.

The co-efficient of heat transmission k_R can still be determined from experimental values, considering a heated passenger carriage as a heating battery and watching the variation in internal air temperature during a determined cooling time z. Based on the dissipation of heat during a unit of time dz :

$$dQ_o = F \cdot k_R\,(t_i - t_{am}) \cdot dz \quad . . . . \quad (4)$$

and the temperature drop dt in the air of the carriage, which is equal to :

$$dQ_o = V_L \cdot \gamma_L \cdot c_L \cdot dt \quad . . . . \quad (5)$$

an equation in which :

V_L = volume of the coach in m³,

γ_L = specific weight of air in kg/m³, and

c_L = its specific heat in kcal/kg °C,

we obtain by combining equations (3) and (4) the differential equation :

$$\frac{dt}{t - t_{am}} = -\frac{1}{\gamma_L \cdot c_L} \cdot k \cdot \frac{F}{V_L} \cdot dz \qquad (6)$$

and by integration :

$$\tau = \frac{t - t_{am}}{t_o - t_{am}} =$$

$$= \exp. -\frac{1}{\gamma_L \cdot c_L} \cdot k \cdot \frac{F}{V_L} \cdot z - K \qquad (7)$$

t being the air temperature at the time z and t_0 the temperature of the air at the time $z = 0$.

With the help of numerous test results obtained on the Deutsche Reichsbahn, and particularly with the aid of temperature readings established by PUTZE [6], taken on trials with AB4 *ü*, C4 *ü*, B4 *i* and C4 *i* built between 1928 and 1931 [7] and [8], we can by using equation (7) calculate the co-efficients of heat transmission k and use them as standards for the determination of the calorific requirements of other types of carriages of similar construction. By averaging the characteristic sizes of passenger carriages designed by the Reichbahn (Table 1), we have calculated the co-efficients of heat transmission k shown in Table 2, using for the air the constants γ_L (kg/m³ and c_L (kcal/kg °C) taken from

TABLE 1. — **Characteristics of passenger carriages.**

Type of carriage	C4 *i*	B4 *i*	C4 *ü*	AB4 *ü*
Type of construction	all steel	all steel	all steel	all steel
Year of construction	1929/31	1929/31	1928/31	1928/31
Length over buffers (mm)	20 960/27 700	20 960/21 700	21 720	21 720
Width of body (mm)	—	—	2 933	2 933
External area of coach (m²)	203	203	204	204
Volume of coach (m³)	138	137	139	138

table D, *b*, 4, of the VDI heat transfer calculator [9].

For determining the calorific needs of the all-steel type coaches now used, we can therefore use the average co-efficient of heat transmission $k_R = 0.93$ to 1.06 kcal/m² h which corresponds very closely to the value mentioned by WINOKUROW for Soviet carriages.

2.2. *Addition* Z_H *for direction.*

With regard to fixed additions in the standard DIN 4701 [10], Table 2, *c*, for direction, Z_H, it must be borne in mind that a passenger carriage is mobile and must take the maximum addition. They must therefore have an increase of :

$$Z_H = 0.05.$$

2.3. *Addition* Z_W *for wind.*

With regard to wind, DIN 4701 distinguishes between normal areas and those with violent winds. Germany is considered to be an area of violent winds in the Zone situated between the North Sea and the Baltic and a line passing south of Osnabrück, Celle, Wittenberge and Schneidemühl. As passenger carriages have no protection from wind, we use the addition :

$$Z_W = 0.25.$$

2.4. *Addition* Z_G *for speed of the train.*

The losses of heat determined whilst standing increase as the speed of travel increases. Supplementary heat losses are caused largely because of convexion, increased by wind when running and by losses due to air leakages, which increase with the speed of the train. To study the influence of speed on heat losses, we will consider an ideal vehicle in which the losses due to leakage defects is nil. The heat losses resulting from the different ventilation fittings and natural air outlets will be taken into account by having an addition for air leakage.

If, in consequence, we consider the ideal carriage placed in a current of air corresponding to the speed of running (m/s), we may express, for the transfer of heat between the air and a plain carriage wall, according to JÜRGES (FALTIN [11], p. 354) :

$$N_u = 0.0568 \cdot R_e^{0.78} \cdot P_r^{0.78} \quad . \; . \; . \; . \quad (8)$$

R_e representing the Reynolds number and P_r the Prandtl number.

By means of the characteristic figures, we get :

$$\alpha_a = 0.0568 \frac{\lambda}{l} \left(\frac{w}{\delta}\right)^{0.78} l^{\,0.78} \left(\frac{\delta \cdot 3\,600}{a}\right)^{0.78} \left[\frac{kcal}{m^2\ h\ {}^oC}\right] \quad . \; . \quad (9)$$

Introducing the constants relative to air, we get :

$$\alpha_a = 0.0429 \frac{\lambda}{l^{\,0.22}} \left(\frac{w}{\delta}\right)^{0.78} \left[\frac{\text{kcal}}{\text{m}^2\ \text{h}\ {}^o\text{C}}\right] \quad . \; . \quad (10)$$

or

$$\alpha_a = 5.66 \frac{w^{0.78}}{l^{0.22}} \; [w \text{ in m/sec}] \quad . \; . \quad (11)$$

Introducing :

$$W = \frac{V}{3.6} \text{ (m/sec)},$$

it becomes :

$$\alpha_a = 2.08 \frac{V^{0.78}}{l^{0.22}} \; [\text{kcal/m}^2\text{h}\ {}^o\text{C}] \; . \; . \; . \quad (12)$$

From DUBBEL [12], p. 440, we can tabulate : for wind speeds lower than 5 m/sec, or 18 km/h, on a co-efficient of thermal exchange of $\alpha = 5.3 + 3.6\, w$ so that for $W = 0$ this co-efficient is equal to $\alpha = 5.3$ kcal/m² h °C. Taking into account the co-efficient of thermal exchange for still air, we can replace equation [12] by the simplified equation :

$$\alpha_a = \frac{0.8\ (V + 10)}{l^{0.22}} \; [\text{kcal/m}^2\text{h}\ {}^o\text{C}] \qquad (13)$$

TABLE 2. — Co-efficients of heat transmission for all steel carriages (*).

Time (h)	C4 *i*					B4 *i*					C4 *ü*					AB4 *ü*				
	t_a [°C]	$t_{ü}$ [°C]	τ_z	K	k [kcal/m²h°C]	t_a [°C]	$t_{ü}$ [°C]	τ	K	k [kcal/m²h°C]	t_a [°C]	$t_{ü}$ [°C]	τ	K	k [kcal/m²h°C]	t_a [°C]	$t_{ü}$ [°C]	τ	K	k [kcal/m²h°C]
0	16.0	48.2	—	—	—	16.5	34.5	—	—	—	16.0	39.6	—	—	—	21.0	36.0	—	—	—
1	16.0	38.8	0.804	0.219	0.794	17.0	27.5	0.798	0.225	0.770	16.0	31.8	0.804	0.218	0.816	21.5	28.4	0.790	0.235	0.751
2	16.0	30.2	0.779	0.250	0.714	17.5	22.0	0.801	0.221	0.831	15.5	27.1	0.854	0.157	1.151	22.0	22.9	0.806	0.215	0.840
3	16.3	23.9	0.792	0.231	0.795	18.0	18.0	0.818	0.200	0.916	15.5	22.7	0.838	0.176	1.041	22.5	18.7	0.817	0.202	0.905
4	16.3	19.4	0.810	0.210	0.875	18.0	15.5	0.862	0.148	1.310	16.0	18.8	0.829	0.186	1.024	23.0	15.6	0.835	0.180	1.020
5	16.3	15.8	0.815	0.203	0.948	18.5	12.0	0.775	0.255	0.752	17.0	15.0	0.799	0.223	0.850	23.5	13.1	0.840	0.173	1.065
6	16.2	12.9	0.816	0.202	0.952	19.5	9.5	0.792	0.233	0.820	17.0	13.0	0.867	0.142	1.338	24.0	10.8	0.825	0.191	0.950
7	16.2	10.6	0.822	0.192	1.001	20.0	7.5	0.790	0.235	0.815	17.0	11.0	0.846	0.166	1.138	24.5	8.9	0.824	0.192	0.989
8	16.0	8.8	0.830	0.185	1.060	20.0	6.0	0.800	0.222	0.880						25.2	7.3	0.820	0.197	0.976
9	16.0	7.5	0.852	0.160	1.250	20.0	5.2	0.866	0.144	1.360						25.5	6.0	0.822	0.196	0.987
				Average :	0.932				Average :	0.938				Average :	1.050				Average :	0.941

(*) The measured values t_a and $t_{ü}$ are taken from tests on the Deutsche Reichsbahn [5].

In addition to the loss of heat due to thermal exchange by convexion, there is a loss by radiation, which is taken into account by the co-efficient of thermal exchange by radiation α_s;

α_s is taken from the equation :

$$\alpha_s = \beta \, C \text{ [kcal/m}^2\text{h °C]} \quad . \; . \; . \quad (14)$$

β being the co-efficient of temperature, and C the co-efficient of radiation.

With :

$\beta = 0.814$ (°C)^3 and C = about 3.8 kcal/m^{2}h (°C)^4,

we get :

α_s = about 3.1 [kcal/m^2 h °C].

The total co-efficient of thermal exchange thus becomes equal to :

$$\alpha_a = \frac{3.1 + 0.8 \, (V + 10)}{l^{0.22}} \text{ [kcal/m}^2\text{h °C]} \quad . \; . \; . \quad (15)$$

$$\alpha_a = \frac{11.1 + 0.8 \, V}{l^{0.22}} \text{ [kcal/m}^2\text{h °C]} \qquad (15a)$$

For a constant difference of temperature $(t_i - t_a)$ we can determine the supplementary calorific requirements resulting from the speed of the train, starting with the co-efficient of heat transmission k. We have :

$$Q = k \, F \cdot \Delta t = f(k) \text{ [kcal/h]}.$$

The co-efficient of heat transmission k from the carriage wall, which is made up of several layers, meets the equation :

$$\frac{1}{k} = \frac{1}{\alpha_i} + \frac{1}{\alpha_a} + \sum_{c=1}^{c=n} \frac{S_n}{\lambda_n} \text{ [m}^2\text{h °C/kcal]} \quad . \; . \quad (16)$$

from which :

$$\frac{1}{\alpha_a} = \frac{1}{k} - \left(\frac{1}{\alpha_i} + \sum_{c=1}^{c=n} \frac{S_n}{\lambda_n} \right) \qquad (16a)$$

At a speed V = 0 km/h and with a co-efficient of heat transmission of k = about 1 kcal/m^2 h °C, determined in paragraph 2.1, it becomes :

$$\frac{1}{5.75} = 1 - 0.826$$

and consequently :

$$\frac{1}{k} = \frac{1}{\alpha_a} + 0.826 \text{ [m}^2\text{h °C/kcal]} \qquad (16b)$$

The co-efficients k determined in function with the speed of the train and the additions Z_G which result are shown in Table 3.

In figure 1, we have traced the curve of the addition Z_G in function with the speed of running. This curve can be expressed by the summarised equation :

$$Z_G = 0.1 \left(\frac{V}{10} \right)^{0.682} - 0.003 \, V \quad . \; . \; . \quad (17)$$

which provides the facility of calculating the necessary addition Z_G for any speed of running.

2.5. *Addition Z_U for air leakages.*

The greatest losses are caused by natural leakages at windows, doors, ventilators, etc., and by the necessary air renewal. If we

TABLE 3. — **Additions Z_G, for running speed.**

Speed (km/h)	0	30	60	90	120
Co-efficient of heat exchange	5.75	18.15	30.6	43.0	55.4
Co-efficient of heat transmission	1	1.13	1.16	1.176	1.185
Addition Z_G	0	0.13	0.16	0.176	0.185

take as l the amount of air changed per hour, we get the loss of heat starting from the specific weight of the air γ_L [kg/m³], the specific heat of the air c_p [kcal/kgm] and the temperature difference $t_{ü}$, by means of the equation :

$$Q_{vu} = l\, c_p\, t_{ü}\, \mathcal{T} \text{ [kcal/h]} \quad . . . . . \quad (18)$$

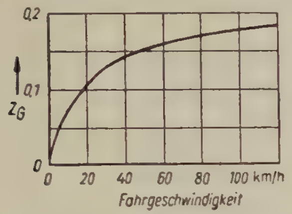


Fig. 1. — Addition Z_G.
N. B. — Fahrgeschwindigkeit = running speed.

By analogy with the method of calculating heat losses following intake of cold air, currently used for heating installations in buildings, the amount of air entering per hour is obtained, according to CAMMERER (W. KÜSTER [13], p. 126) by the equation :

$$V_h = L\, a\, (p_a - p_i)^{2/3} \text{ [m}^3\text{ h]} \quad . . . \quad (19)$$

in which :

L = length of joints, leaks, etc., in m,

a = constant for quality of windows, doors, etc., and

$(p_a - p_i)$ = difference of pressure between inner and outer faces, in mm of water.

During the running of the train at the speed V, the kinetic energy of the air encountered is converted to pressure and this is equal to :

$$p_{st} = \left(\frac{V}{3.6}\right)^2 \cdot \frac{\gamma L}{2\, g} \text{ [mm water]} \quad . . \quad (20)$$

Because of the pressure, a portion of the air enters the carriage and there causes an increase in pressure above the value p_i, this increase depending on the speed with which the air is subsequently released from the carriage. For passenger carriages, we can, as in the sections comprising opposite windows, take as variation of the effective pressure :

$$p_a - p_i = 1/2\, P_{st} \quad . . . . \quad (21)$$

By this fact, we take from (19) :

$$V_h = L\, a \left[\frac{1}{2}\left(\frac{V}{3.6}\right)^2 \frac{\gamma L}{2g}\right]^{2/3} \text{ [m}^3\text{/h]} \quad (22)$$

and with :

$$\frac{\gamma L}{g} = 0.127,$$

we get :

$$V_h = L\, a \cdot 0.0188\, V^{1.33} \quad . . . . . \quad (22a)$$

Replacing L, length of joints, leaks, etc., and the constant a in equation (22a) by a proportional size μ and the volume of air I in the carriage, we can express generally :

$$V_h = \mu\, I\, V^n \text{ [m}^3\text{/h]} \quad . . . . \quad (22b)$$

The amount per hour of air entering the carriage, V_h, must be proportional to the required amount of replacement air. Introducing a co-efficient l_{wv} for renewal of air which indicates the number of times 1 m³ of air is renewed per hour, the amount of air changed for the whole of the carriage with an air volume 1 (m³) is raised to :

$$V_h = l_{wv}\, I \text{ [m}^3\text{/h]} \quad . . . . \quad (23)$$

Balancing the two equations, we get a co-efficient of air renewal at speed V :

$$l_{wv} = \mu V^n \text{ [h}^{-1}\text{]} \quad . . . . \quad (24)$$

The air renewal shown has been achieved by the speed of running V; it is however also effective at a speed V = 0 km/h and is then produced by the temperature difference $t_i - t_a$. If the internal and external temperatures are equal, there will be no air change, because $t_{ü} = t_i - t_a = 0$. The more the temperature varies the greater the amount of air changed per cubic metre, so that we can say :

$$l_{tü} = \lambda\, (t_i - t_a) \text{ [h}^{-1}\text{]} \quad . . . . . . \quad (25)$$

Total renewal of air per cubic metre caused by the speed V and the difference of temperature $t_i - t_a$ thus becomes :

$$l = l_{wv} + l_{t\ddot{u}} \; [\mathrm{h}^{-1}] \quad \ldots \ldots \quad (26)$$

$$l = \mu \, V^n + \lambda \, (t_i - t_a) \; [\mathrm{h}^{-1}] \quad \ldots \quad (26a)$$

The co-efficients μ and λ used are easily determined from the well-known test results on the Deutsche Reichsbahn [6]. On the basis of the details given by PUTZE in his article on « Tests of passenger carriage heating » [6], we arrive at the values shown in Table 4 for μ and λ, which can be used to determine the calorific needs of ordinary passenger carriages. Taking a value $n = 0.95$ for the index n, we come very close to the Deutsche Reichsbahn test results.

We can thus in a general way calculate by equation (18) the heat loss resulting from leakage and the necessary air change; by means of the ratio :

$$\frac{Q_{vu}}{Q_o} = Z_L \quad \ldots \ldots \ldots \quad (27)$$

We can then determine the addition Z_L from the amount of heat necessary when at a stand with no leakage losses.

We have :

$$Z_L = \frac{[\lambda \, (t_i - t_a) + \mu V^{0.95}] \, c_p \gamma_L \, (t_i - t_a) J}{k_R \, F (t_i - t_a)} \quad (28)$$

and if we take $\gamma_L = 1.297$ kg/m³ and $c_p = 0.24$ kcal/kg as an average value, it becomes :

$$Z_L = \frac{[\lambda t_{\ddot{u}} + \mu V^{0.95}] \cdot J \cdot 0.31}{k_R F} \quad \ldots \quad (29)$$

or, with k = about 1 kcal/m² h,

$$Z_L = \frac{J}{F} 0.31 \, [\lambda t_{\ddot{u}} + \mu V^{0.95}] \quad \ldots \quad (30)$$

With this equation we can then, knowing the volume of air contained in the carriage I [m³] the total external area of the carriage F [m²] and the values of λ and μ shown in Table 4, determine the extra calorific energy subsequently necessary for the renewal of air in relation to the temperature difference $t_{\ddot{u}}$ and the speed V.

TABLE 4. — **Values of λ and μ.**

	λ	μ
Natural leakage	0.040	0.08
Ventilator closed (30 %)	0.054	0.102
One shutter opened	0.0780	0.118
Ventilator opened (100 %)	0.0780	0.142
Two shutters opened	0.120	0.22

2.6. *Formula for calculating total calorific needs.*

If we introduce the additions found by the initial equation (1) for calorific needs, we get :

$$Q_t = k \, F t_u \cdot [1.3 + 0.31 \frac{J}{F} (\lambda \, t_{\ddot{u}} + \mu V^{0.95}) + 0.0235 \, V^{0.682} - 0.003 \, V] \; [\mathrm{kcal/h}] \quad (31)$$

In this formula :

k = co-efficient of heat transmission in kcal/m² h °C (from 2.1, k is about 1),

F = total external area in m² of the carriage,

J = colume of the carriage in m³,

$t_{\ddot{u}}$ = difference of temperature $t_i - t_a$,

V = speed in km/h,

λ = co-efficient from Table 4, and

μ = co-efficient from Table 4.

In figures 2 and 3 are shown the calorific requirements for an express AB4*ü* in coach built in 1932 from the equation (31) for different ventilation conditions. Test results from the Deutsche Reichbahn [6] recorded with a vehicle of the same type show close agreement between the calculated and

measured needs. To continue the investigation we can then calculate the calorific requirements of the all-steel vehicles of present day types by the equation set up (31). If we admit that during the heating period, the ventilators are closed, we get the values of calorific needs shown in figure 4 as a function of the difference in temperature $t_i - t_a$ and the speed V.

by the expression :

$$Q_v = \varphi Q_i n \text{ [kcal/h]} \quad . \quad . \quad . \quad . \quad . \quad (33)$$

In Table 5, we have shown the heat losses in the various heating systems to provide for determination of the heating index. For this, we have taken as a basis a ten coach set because longer trains are rarely operated in Germany.

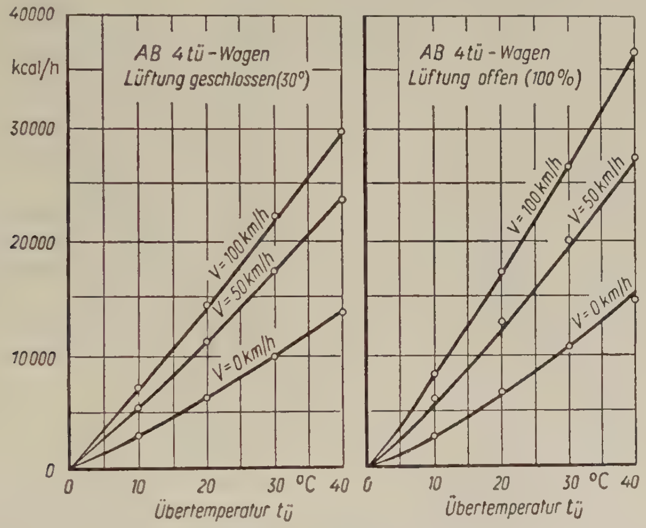


Fig. 2 and 3. — Calorific requirements of carriages, 1932 type of construction.

N. B. — Lüftung geschlossen = ventilators closed. — Lüftung offen = ventilators open. — Übertemperatur = temperature difference $t_ü$.

3. Comsumption of heat for heating a carriage or a train.

Because of losses in the heating system, for example incomplete combustion in the firebox or boiler, losses of steam at couplings, etc., and grouping them all to give an overall efficiency, η_{Hzg}, the consumption of heat for a vehicle is expressed by :

$$Q_v = Q_i \cdot \frac{1}{\eta_{Hzg}} \cdot n \text{ [kcal/h]} \quad . \quad . \quad . \quad . \quad (32)$$

or making use of a « heating index » :

$$\varphi = \frac{1}{\eta_{Hzg}}$$

Knowing the heating index and calorific needs Q_i, we also know the hourly consumption of heat for the possible methods of heating trains in service, apart from steam-heating with crude oil fired boilers and exhaust gas. The consumption of heat for a 10 coach passenger train is shown graphically in figure 5.

3.1. *Consumption of heat when using exhaust gas or crude oil fired boiler.*

Consequent on the use of exhaust gas from Diesel motors, the consumption of heat will clearly be less than with an ordinary crude oil boiler. For comparison,

we will examine a Diesel locomotive of 1 800 to 2 000 HP with two high speed supercharged motors.

3.2. *Use of exhaust gas heat from the Diesel motors.*

From the load of the 900 HP Diesel motor at 1 500 r.p.m. for example, such as is designed for the Deutsche Reichsbahn locomotives, we get the valves shown in Table 6 and figures 6 and 7 in which use has been made in part of best results from an English Diesel motor of 12 cylinders and 910 HP [14].

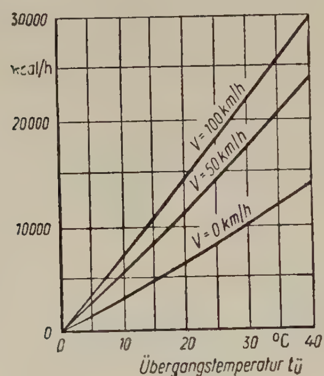


Fig. 4. — Calorific requirements of all steel carriages (30 % ventilation and shutters closed).

3.3. *Power required from the blower.*

When the exhaust gas from a supercharged Diesel motor is taken to a boiler heated by means of this gas, there is set up, in the exhaust pipe, as a result of resistance of the boiler, h_1, a counter pressure which reduces the power of the turbo-compressor. It is therefore necessary either to provide a more powerful turbine or to compensate the boiler resistance h_1 by means of a fan at the intake to the boiler. In the equipment designed by the Rolling Stock Institute, the latter arrangement has been used. The power required from the fan is shown by the equation :

$$N = \frac{Q\gamma h_1}{75\ \eta}\ [\text{HP}] \quad \ldots\ldots\ldots \quad (34)$$

in which :

Q = total volume of gas in m^3/h,

γ = specific weight of exhaust gas in kg/m^3, and

η = overall efficiency.

We get the fan powers given in Table 7 and figure 8 in relation to the load of the motors and the resistance of the boiler.

The exhaust gas and crude oil boiler equipment designed for the 1 800 HP Diesel locomotive has a resistance of about 300 mm water so that, deducting the power needed for the fan, we get the amount of heat actually usable shown in figure 7.

On the basis of the curves showing hourly consumption of heat in relation to the difference of temperature $t_{ü}$, speed V and method of heating (fig. 5), we get the case in which the additional heat must be used, in other words where there is additional consumption of Diesel fuel.

From the expression for power of the motors :

$$N = \frac{ZV}{270\ \eta}\ [\text{HP}] \quad \ldots\ldots\ldots \quad (35)$$

we get, for the speed V [km/h] and the tractive effort required at this speed Z [kg], the powers shown in Table 8 for the transport of the 10 coach set under consideration.

The resistance to movement of the locomotive and carriages have been taken from

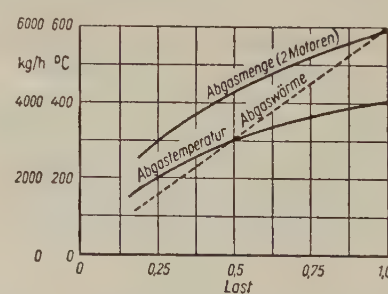


Fig. 6. — Temperature, quantity and total amount of exhaust gas from Diesel motors.

N. B. — Abgasmenge = amount of exhaust gas. — Abgastemperatur = temperature of exhaust gas. — Abgaswärme = heat content of exhaust gas.

TABLE 5. — Losses (in kcal/h) and consumption of heat for train heating.

No.	Designation	Method of heating							
		Steam heating. Oil fired boiler on locomotive	Hot water heating				Heat exchanger		Electric resistance heating
			Boiler on coach		Boiler on locomotive				
			Oil-fired boiler	Coal boiler	Oil-fired boiler	Coal boiler	Steam hot water	Electric hot water	
1	Condensation in train pipe	4					4		
2	Condensation on the coach	11							
3	Escape and losses from train pipe	43					43		
4	Unburnt firebox residue			5					
5	Inadequate combustion		2						
6	Ash and soot			3					
7	Unburnt gas		1	2					
8	Free heat in discharged gas		10	10					
9	Radiation in the heater		10	10			5	5	
10	Losses in couplings and pipe				20	20			
11	Loss of load along the train							2.5	2.5
12	Total losses	58	23	30	20	20	52	7.5	2.5
13	Efficiency of heating from locomotive or boiler van %	42	77	70	80	80	48	92.5	97.5
14	Boiler efficiency %	70			77	70	70		
15	Diesel motor efficiency %							32	32
16	Generator efficiency %							97	97
17	Overall efficiency %	29.4	77	70	61.6	56	33.6	28.7	30.3
18	Heating index φ	3.4	1.3	1.43	1.62	1.78	2.98	3.48	3.3

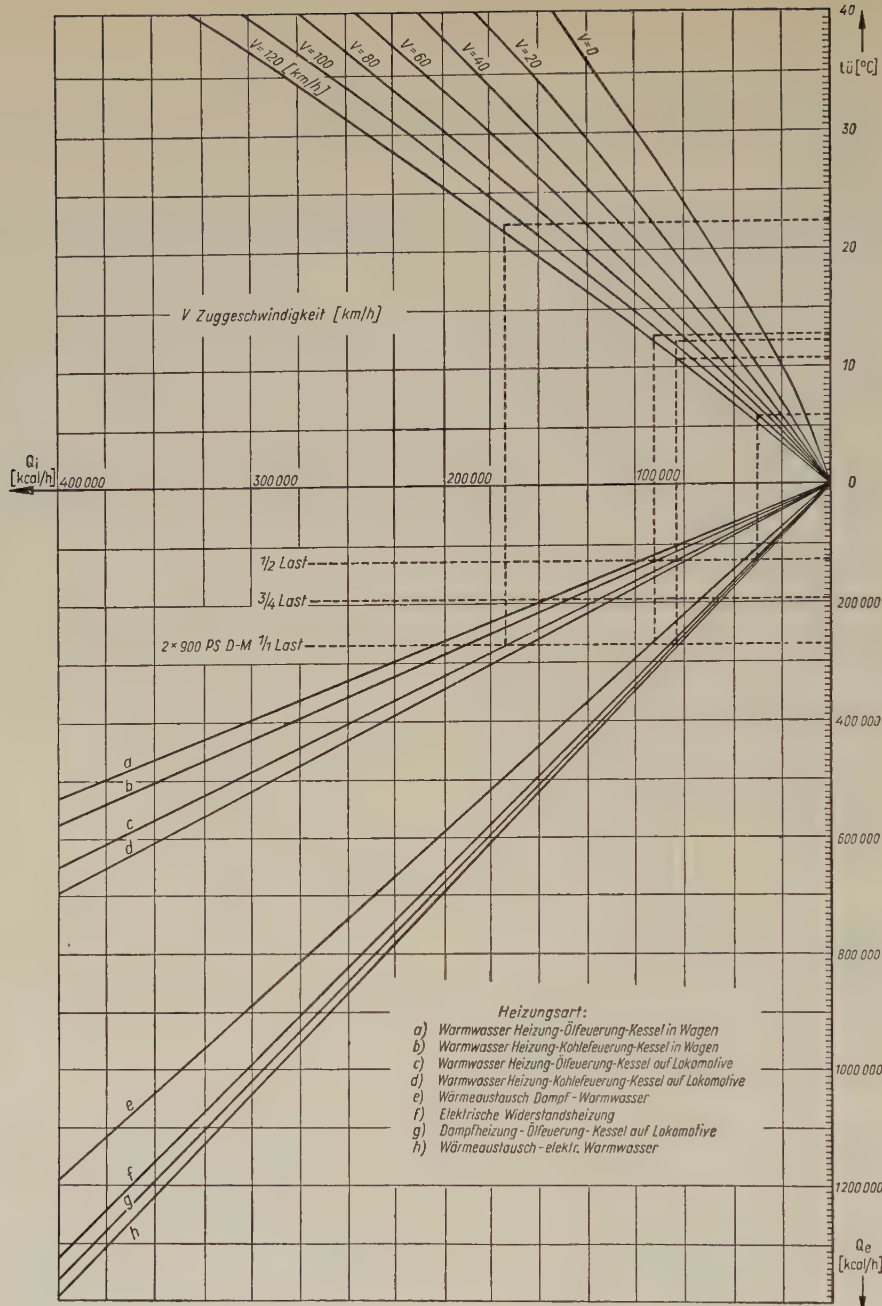


Fig. 5. — Consumption of heat for a ten coach passenger train.

($t_{ü} = ti - t_a$; $ti = 20°C$).

N. B. — Last = load. — Zuggeschwindigkeit = running speed.

Translation of the German terms :

Type of heating :

a) Hot water heating, oil fired boiler on the carriage;
b) Hot water heating, coal fired boiler on the carriage;
c) Hot water heating, oil fired boiler on the locomotive;
d) Hot water heating, coal fired boiler on the locomotive;
e) Hot water/steam heat exchanger;
f) Electric resistance heating;
g) Steam heating : oil fired boiler on the locomotive;
h) Hot water/electric heat exchanger.

TABLE 6. — **Use of Diesel exhaust gas.**

Motor rating	HP	225	450	675	900
Temperature of exhaust gas beyond the turbine	°C	200	300	350	400
Enthalpy of exhaust gases (1)	kcal/kg	50	76	88	102
Co-efficient of excess air (2)	—	2.3	1.8	1.5	1.3
Specific consumption of fuel (2)	g/HP.h	200	190	180	180
Theoretical air requirement L_{min} (approx.)	kg/kg	14	14	14	14
Quantity of air L_{tat}	kg/HP. h	6.45	4.8	3.78	3.28
Volume of exhaust gases ($V_g L_{tat}$)	kg/h	1 450	2 160	2 550	2 950
Heat contained in exhaust gas (one motor)	kcal/h	72 500	164 000	224 000	300 000
Heat contained in exhaust gas (two motors)	kcal/h	145 000	328 000	448 000	600 000
Temperature of exhaust gas leaving the boiler (3)	°C	130	190	200	230
Difference of temperature t	—	70	110	150	170
Enthalpy of gas leaving the boiler (1)	kcal/kg	33	47	50	56
Amount of heat used	%	34	38.2	43.2	45
Amount of heat used	kcal/h	49 400	125 000	193 000	270 000
Possible quantity of steam	kg/h	75	190	294	412

(1) From enthalpy figures by W. GUMZ [15].
(2) From workshop figures from Johannisthal Works, Motor Suppliers.
(3) Values found by Hohenthurm Boiler Works.

TABLE 7. — **Power needed at the blower.**

Motor rating	225	450	675	900	HP
Volume of exhaust gases (two motors)	2 250	3 350	3 960	4 580	N m³/h
Additional amount of oil needed to reach maximum production of steam of 780 kg/h	70.5	59	48.6	36.8	kg/h
Volume of exhaust gases (L_{min} = 10.22 N m³/kg; λ = 1.3)	940	785	646	490	N m³/h
Volume of exhaust gases Q total	3 190	4 135	4 506	5 070	N m³/h
N = (η = 0.8) h_1 = 100	1.90	2.47	2.70	3.03	HP
= 200	3.80	4.94	5.40	6.06	HP
= 300	5.70	7.31	8.10	9.09	HP

TABLE 8. — **Power needed by the locomotive to haul a train of ten coaches on level track.**

Speed	V	40	60	80	100	120	km/h
Tractive effort	Z	1 360	1 760	2 370	3 080	3 740	kg
	ZV	54 400	105 800	189 500	308 000	449 000	—
Power	N	260	490	880	1 430	2 080	HP

the Büttner [16] calculation charts and we have taken as the average efficiency $\eta = 0.8$.

With the possible speed and power, we can determine on figure 5 the temperature

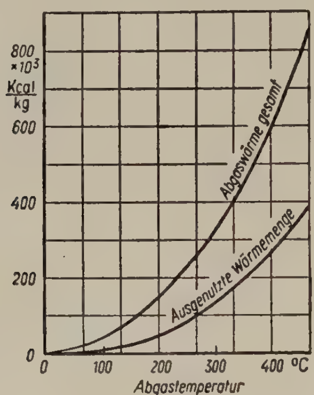


Fig. 7. — Quantity of heat and usable quantity of heat.

N. B. — Abgaswärme gesamt = total heat of exhaut gas. — Ausgenutzte Wärmemenge = usable amount of heat. — Abgastemperatur = temperature of exhaut gas.

zones in which heating can be ensured solely by the exhaust gas from the Diesel motors, without the help of oil heating.

In all other conditions of operation, the consumption of heat is given by :

$$Q_{AH} = Q_e - Q_A \text{ Diesel [kcal/h]} \qquad (36)$$

Q_e can be noted on figure 5 and the amounts of usable heat from the exhaust gas of the Diesel motors, Q_A Diesel are given in Table 6.

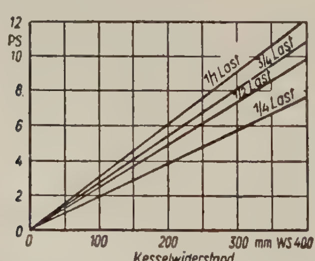


Fig. 8. — Power required for exhaust gas blower.

N. B. — Kesselwiderstand = boiler resistance.

4. Usage factor of a train heating installation.

4.1. *Distances run by a Diesel locomotive.*

From the statistical data of the O.R.E. (Research and Tests Office of the International Railway Union), the average distance run by a Diesel train locomotive in Western Europe is about 45 000 km/annum whilst in America 225 700 km/annum are achieved. From the report of the « Dieselschienenverkehr » (Diesel Traffic Commission) [18], p. 99, the annual operation of Diesel locomotive in Germany is about 3 000 h average. To take account of maintenance and examination, 340 days per annum can be taken as operation so that the daily period of operation for a line locomotive is :

$$l_T = \frac{3\,000}{340} = 9 \text{ hours per day}$$

TABLE 9. — **Degree-days (*) and days of heating in the German Democratic Republic.**

Town	*Degree days*	*Days of heating*
Karl-Marx-Stadt . .	3 570	237
Erfurt	3 510	233
Berlin	3 420	226
Leipzig	3 290	222
Halle	3 260	226
Magdebourg	3 240	220
Baltic Region . . .	3 600	247
Total . . .	23 890	1 611
Average for the German Republic. . .	3 420	230

(*) Degree-days = days of heating multiplied by the average difference between internal and external temperatures.

TABLE 10. — **Average speeds of some Deutsche Reichsbahn express trains.**

Train number	*Journey*	*Length of line km*	*Time for journey*	*Average speed km/h*
D 32	Berlin — Saalfeld	293	6 h 54 min	42.5
D 152	Berlin — Saalfeld	293	6 h 6 min	48.0
Dt 34	Berlin — Leipzig	164	3 h 11 min	51.4
D 30	Berlin — Leipzig	164	3 h 33 min	46.8
D 186	Berlin — Plauen	286	7 h 18 min	39.2
D 185	Plauen — Berlin	286	5 h 53 min	48.5
D 120	Dresden — Plauen	177	3 h 50 min	45.5
D 105	Plauen — Dresden	177	3 h 54 min	45.4
D 96	Dresden — Gera — Erfurt	289	5 h 54 min	49.1
D 198	Leipzig — Erfurt	177	1 h 53 min	61.6
D 84	Dresden — Riesa — Leipzig	120	2 h 23 min	50.0
D 82	Leipzig — Magdeburg	119	2 h 12 min	54.2
D 86	Wittenberge — Magdeburg	112	2 h 20 min	47.6
D 183	Wittenberge — Magdeburg	112	1 h 51 min	59.4
D 108	Dresden — Leipzig	120	2 h 22 min	50.0
D 83	Leipzig — Dresden	120	2 h 16 min	53.2
D 81	Leipzig — Karl-Marx-Stadt	80.6	1 h 44 min	46.8
D 82	Karl-Marx-Stadt — Leipzig	80.6	1 h 31 min	53.6
Average speed of G.D.R. expresses				50.0

4.2. *Degrees-days, days of heating, hours of heating.*

The basis of determination of degree-days of heating and consequently period of use b of a train heating installation is the curve of annual temperature of the country in which the train is mainly in service. From KÜSTER [13] we get for the territory of the German Democratic Republic the days of heating shown in Table 9.

If we allow that of the number of non-working days provided by 4.1. for maintenance and repair, twelve days fall in the heating season, we arrive at a period of use of :

b = 218 annual heating days and 3 280 degree-days.

With 9 hours daily use of the locomotive, the annual number of hours of heating amounts to 1 965 h/annum. The utilisation factor of the train heating equipment :

$$\alpha = \frac{b}{8\,760} \quad \ldots\ldots\ldots\ldots \quad (37)$$

thus amounts, when b = 1 965 h/annum, to α = 0.225.

4.3. *Average speed of running during the heating season.*

As the consumption of heat, and consequently the costs of a train heating equipment depend largely on the speed of traveil V, it is essential to determine the average speed of fast trains over the territory of the German Democratic Republic. According to the official timetable, the average speeds are as shown in Table 10.

From the average speed of running, we can determine the annual cost of heating in relation to degree-hours — hours of heating multiplied by the average difference between interior and exterior temperatures.

5. **Annual cost of heating.**

5.1. *Cost of fuel.*

The hourly cost of fuel is given by :

$$k_b = Q_i \cdot \frac{1}{\eta_{Hzg}} \cdot \frac{p}{10^6} \text{ [DM/h]} \quad \ldots \quad (38)$$

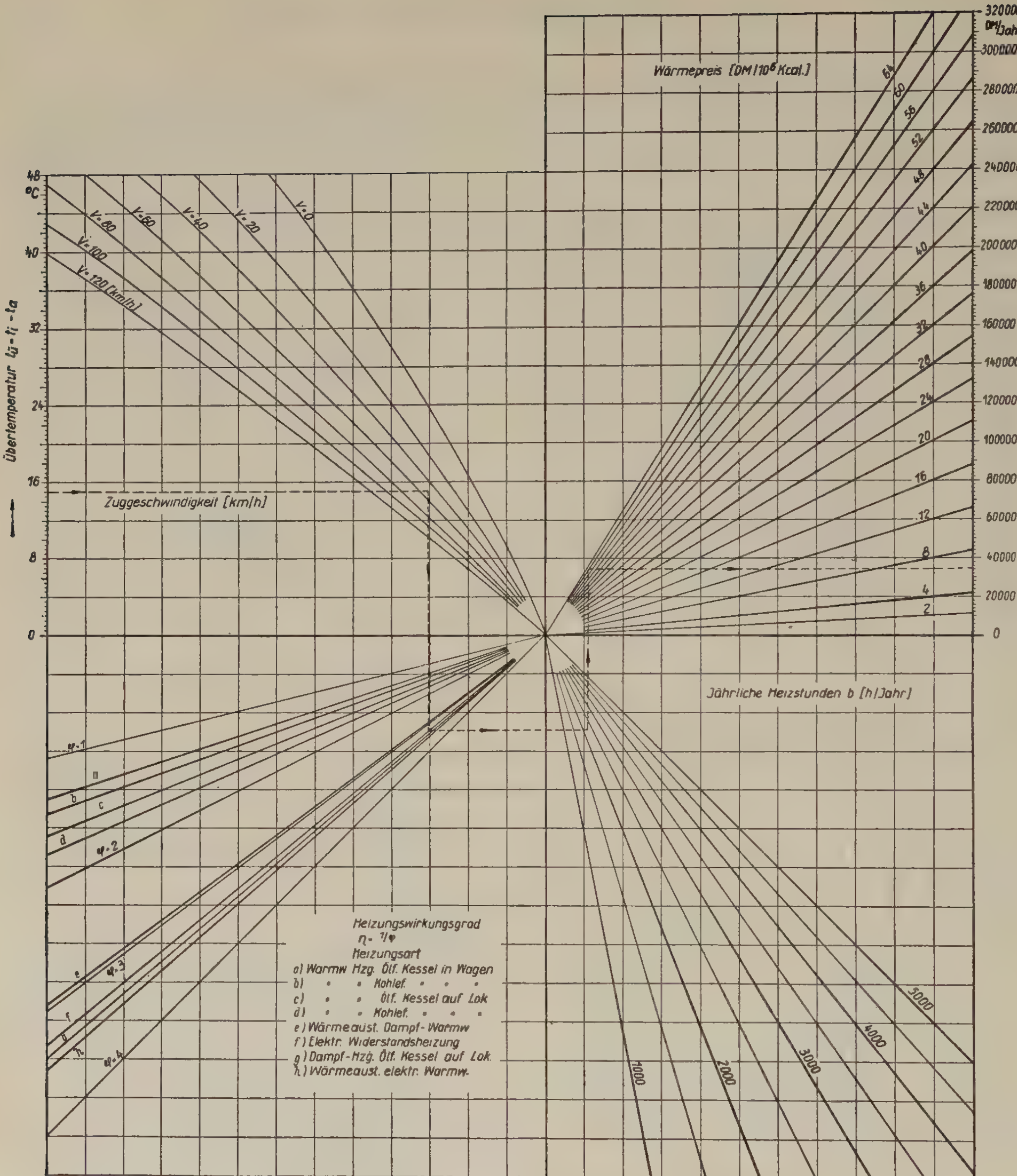


Fig. 9. — Annual cost of heating a passenger train (10 all steel coaches).

N. B. — Übertemperatur = difference of temperature. — Zuggeschwindigkeit = speed of train. — Wärmepreis = cost of one unit of heat. — Jährliche Heizstunden = annual number of heating hours. — Heizungswirkungsgrad = heating efficiency. — Heizuggsart = type of heating.

Example : Difference of temperature $t_{ü}$ = 15° C;
Average speed V = 50 km/h;
Electric resistance heating (Diesel generator);
Annual heating hours b = 2 200 h/annum;
Price per unit of heat of Diesel oil p = 60.70 DM/10⁶ Kcal.

By following the curve, we get an annual heating cost of a ten coach train of K_b = 32 200 DM/annum.

in which :

Q_i = calorific needs in kcal/h from equation 31 or figure 4,

η_{Hzg} = overall efficiency, and

p = price per million kcal in DM.

From the annual hours of heating and degree-hours determined in 4.2 we take the average temperature difference $t_{ü}$.

$$t_{ü} = \frac{\text{degree-hours}}{\text{hours of heating}} \quad (^{\circ}\text{C})$$

which, by equation 31 or figure 4, allows the determination of the calorific needs Q_{im} in kcal/h, so that the annual costs of fuel are shown to be :

$$K_b = Q_{im} \cdot \frac{1}{\eta_{Hzg}} \cdot \frac{p}{10^6} \cdot b \ [\text{DM/an}] \quad (39)$$

From the regulations in force, the price of heating and fuel at present in the German Democratic Republic are as shown in Table 11.

In figure 9 are shown the annual costs of fuel. From the representation of equation (39), we can see the annual cost of fuel for all possible operating conditions, the differences of temperature $t_{ü}$, the speeds V, and any method of heating whatsoever. With regard to the conditions applicable to the German Democratic Republic, we can determine as an example the annual costs of fuel for :

given an express train of 10 all steel carriages, annual hours of heating, b = 1 965 from 4.2, average speed, V_m = 50 km/h we can calculate :

Average difference of temperature $t_{ü}$:

$$t_{ü} = \frac{3\,420 \text{ degree-days}}{230 \text{ days of heating}} = 15^{\circ}\text{C}.$$

On the chart (fig. 9) we see, commencing with $t_{ü}$ = 15° C, V_m = 50 km/h, and taking account of the method of heating, period of use b, and cost per calorie p, the fuel costs shown in Table 12, according to the use either of Diesel oil at 60.7 DM/10^6 kcal, or of lignite briquettes at 3.96 MD/10^6 kcal.

5.2. *Installation costs. Annual charges.*

The central heating equipment of the Diesel locomotive or heating van will be designed to serve a maximum of 10 or 12 carriages. From information supplied by manufacturers, the costs of equipment P in the German Republic amount to :

oil-fired boiler with all accessories, about	60 000 DM
exhaust-gas and oil-fired boiler with all accessories about .	65 000 DM
heat generator including the Diesel motor about . . .	85 000 DM
coal fired boiler about . . .	2 080 DM
fully-automatic oil-fired boiler about	4 000 DM
heat exchanger, complete about	3 000 DM

The annual amount for sinking fund charge is calculated to redeem the capital cost in 15 years. The interest charge is z = 0.07 or 7 %. The annual capital payment is thus :

$$K_k = Pz \ [\text{DM/annum}] \quad (40)$$

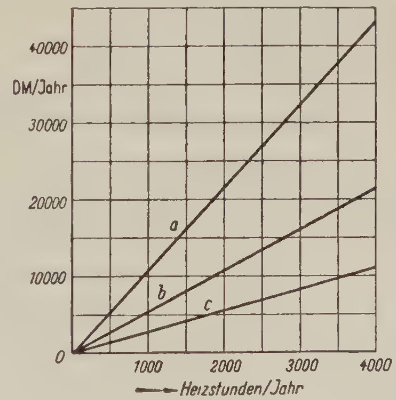


Fig. 10. — Annual cost of operation.

a) Coal heating in ten coaches;
b) Coal heating on the locomotive or in five coaches;
c) Oil heating in five or ten coaches respectively.

	Ten coach set		*Five coach set*
Oil fired boiler		4 200.—	
Exhaust gas and oil-fired boiler		4 550.—	
Heat generator and Diesel motor		5 950.—	
Coal-fired boiler	1 460.—		730.—
Oil-fired boiler	2 810.—		1 405.—
Heat exchanger	2 102.—		1 051.—

TABLE 11. — **Prices of fuel in the German Democratic Republic.**

Fuel	*Type of charge*	DM/*t*	DM/10^6 kcal
Diesel fuel. . . . . . (H_u = 10 000 kcal/kh)	Producer's price	365	36.5
	Addition for the D.R.	240	24.0
	Stocking, etc.	2	0.2
	Total price	607	60.7
Lignite briquettes. . . (H_u = 4 800 kcal/kg)	Producer's price	16.52	3.45
	Increase.	—	—
	Stocking, etc.	2.48	0.51
	Total price	19.00	3.96

TABLE 12. — **Annual cost of fuel for a train of ten or five coaches for 1 965 hours of heating per year and an average temperature difference of $t_{ü}$ = 15° C.**

Type of heating (letters as in fig. 5)	*b*	*d*	*a*	*c*	*e*	*f*	*g*	*h*
Ten coach train, DM/annum . .	520	900	12 000	14 500	27 000	30 000	30 700	31 500
Five coach train, DM/annum . .	260	450	6 000	7 250	13 500	15 000	15 350	17 750

TABLE 13. — **Annual cost of heating for a ten or five coach train for 1 965 hours of heating per year and an average temperature difference of $t_{ü}$ = 15° C.**

Type of heating (letters as in fig. 5)	*d* / *c*	*c* / *b*	*a* / *a*	*b* / *d*	*e*	*g*	*f*	*h*
Ten coach train, DM/annum . .	13 800	17 100	20 050	23 800	33 200	35 000	36 000	38 200
Five coach train, DM/annum . .	11 100	11 100	12 800	13 100	18 700	19 600	21 100	22 700

5.3. *Operating costs.*

By operating costs, we refer to the costs incurred by attending to the central heating boiler or the separate carriage boilers.

In the case of a centralised equipment on a Diesel locomotive, there is, with fully-automatic control, no additional operating cost. With a central coal-fired boiler, we need a fireman for the full heating period *b*. With coal fired boilers on the carriages, a fireman is needed for each five carriages.

Taking an hourly wage rate of 1.80 DM for the fireman and a general overhead percentage of 200 %, the following are the operating costs K_{bd} :

Annual hours heating	1 000	2 000	3 000	4 000
Coal firing, central, DM/annum	5 400	10 800	16 200	21 600
Coal firing in 10 coaches, DM/annum	10 800	21 600	32 400	43 200
Coal firing in 5 coaches, DM/annum	5 400	10 800	16 200	21 600
Oil firing in 5 or 10 coaches, DM/annum	2 700	5 400	8 100	10 800

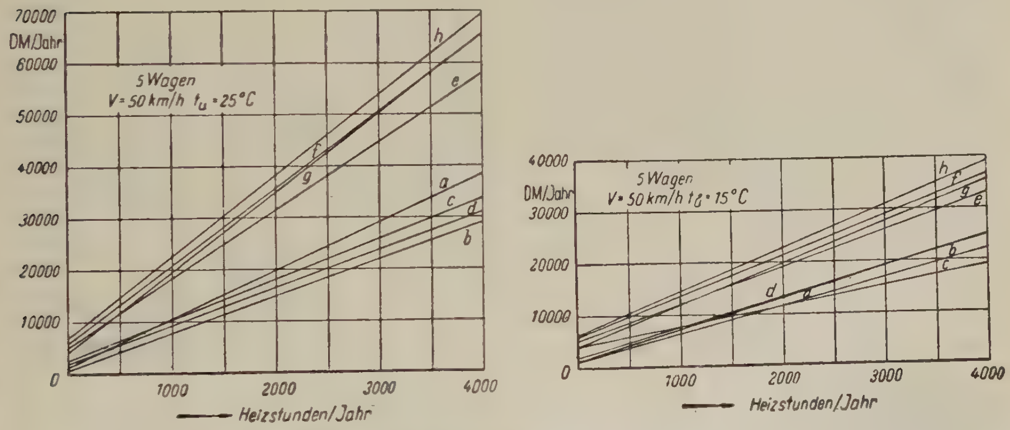


Fig. 11 and 12. — Total annual cost of heating a train of 10 coaches (5 coaches) average speed $V_m = 50$ km/h, temperature difference $t_{ü} = 15^{\circ}$ C.

N. B. — For the explanations relative to the letters shown in these figures, please refer to Fig. 5.

Figure 10 shows the operating costs in relation to the annual number of hours of heating.

5.4. *Total annual costs.*

The total annual costs are given by :

$$K = K_b + K_k + K_{bd} \text{ [DM/an]} \quad . \; . \quad (41)$$

The total annual costs can be determined from figure 9 for all methods of heating and for any number of hours of heating, from 5.2 for the annual capital costs and from figure 10 for the operating costs. In the conditions at present obtaining on the Reichbahn, we arrive at the total costs shown in Table 13 for a train of ten or five carriages with 1 965 annual hours of heating and an average difference of temperature of $t_{ü} = 15^{\circ}$ C.

The total annual heating costs in relation to the number of hours of heating for similar conditions are shown in the figures 11 and 12. The figures 13 and 14 give the heating costs, which would be needed, if all other conditions analogous, the average

difference temperature would reach $t_{ü}$ = 25° C.

In the conditions shown, applicable to the German Reichbahn, it appears from our investigation that hot water heating with a coal-fired central boiler would be the most economical way of heating a train of 10 coaches. Next comes the same method of heating with a central oil-fired oil-fired boiler : the conversion of existing carriages even can be recommended, because the considerably reduced cost of heating justifies such a conversion and the cost of the change over is liquidated in a few years.

From the methods of heating at present used on the Reichbahn, the traditional steam heating has scarcely any advantage

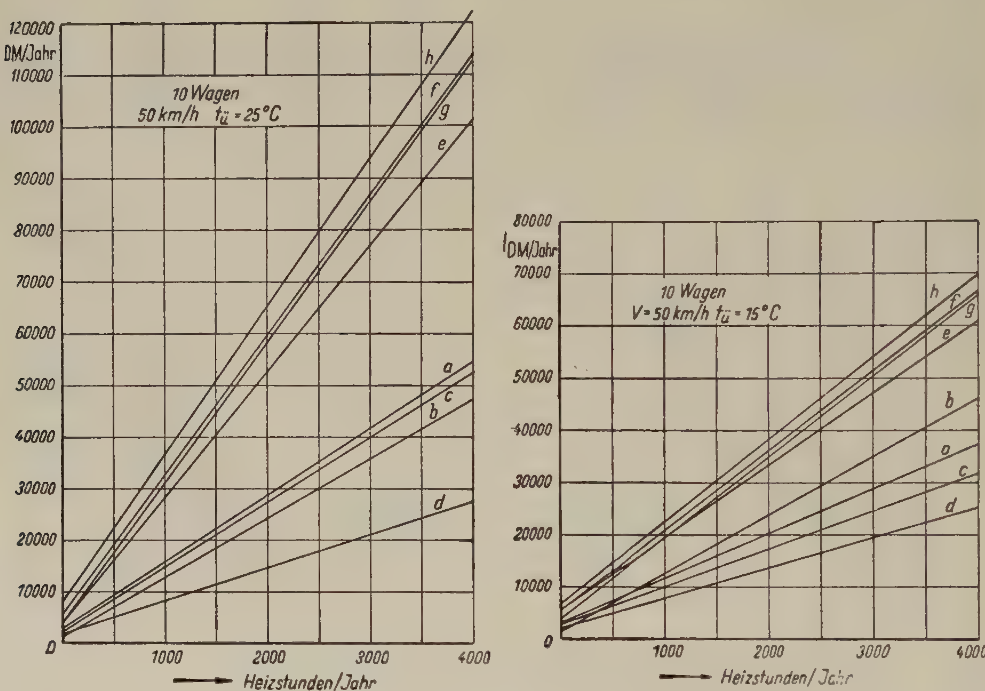


Fig. 13 and 14. — Total annual cost of heating a train of 10 coaches (5 coaches); average speed V_m = 50 km/h, temperature difference $t_{ü}$ = 25° C.

N. B. — For the explanations relative to the letters shown in these figures, please refer to Fig. 5.

boiler and then hot water heating with a separate boiler for each coach. In the diagrams forming figures 11 to 14, we see that hot water heating is economically superior to other methods of heating, even for the 5 coach set and for high rates of supply. Moreover, in the case of all Diesel or all-electric traction, it would be appropriate to provide all newly-built carriages with hot water heating and a fully-automatic over electric resistance heating for a 10-coach set or for a 5-coach set. It is therefore not unusual to use with Diesel traction electric resistance heating with current supplied by a central Diesel-generator set.

Hot water heating for steam or electric traction, designed by Hagenuk of Keil, and fitted on 150 new carriages of the Reichbahn [2] gives good results from

the economic point of view when the heat exchanged is fed with steam.

5.5. *Annual heating costs using exhaust gas and oil-fired boiler.*

In chapter 4, we determined the usage factor of a train heating installation, in the conditions on the Deutsche Reichbahn, we found 1 965 hours of heating per year, 15° C difference of temperature and 50 km/h average running speed.

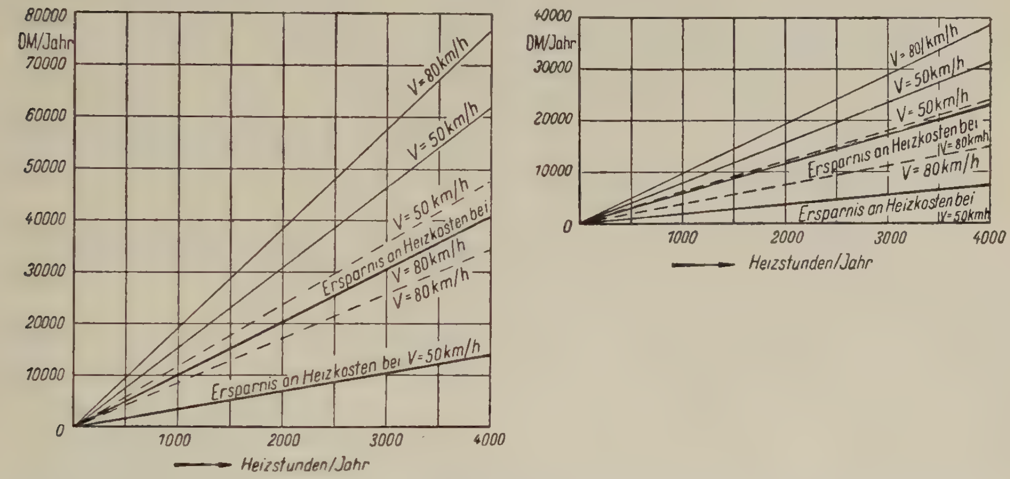


Fig. 15 and 16. — Annual costs of fuel for heating a train of ten or five coaches, with a temperature difference of 15° C, using a central steam boiler, oil fired, or oil-fired and exhaust gas, savings effected by using exhaust gas.

———— Oil fired boiler. .... Oil-fired and exhaust gas boiler.

N. B. — Ersparnis an Heizkosten bei... = Saving in cost of heating at...

From Table 9, the haulage of a train of 10 coaches on level track at a speed of 50 km/h demands an engine power of 370 HP and a train of 5 coaches 195 HP. From the details in Table 6, the quantities of heat amount to :

Q_{a10} = 40 000 kcal/h for the 10 coach set, and

Q_{a5} = 21 000 kcal/h for the 5 coach set.

To allow the practical use of these quantities of heat, it is necessary to consume a quantity of heating oil as follows :

$$G = \frac{Q_a \text{ [kcal/h]}}{10\,000 \text{ [kcal/kg]} \cdot \eta_k} \text{ [kg/h]} \quad . \ . \quad (42)$$

which, with a boiler efficiency of $\eta_k = 0.7$, gives :

G_{10} = 5.72 kg/h of fuel oil, or

G_5 = 3 kg/h of fuel oil.

Each year, therefore, we use an amount of heating oil at least equal to G times the number of heating hours, or in the present case :

for a 10 coach set = 11 220 kg/annum Diesel oil, or 6 820 DM/annum.

for a 5 coach set = 5 900 kg/annum Diesel oil, or 3 580 DM/annum.

After deduction of financial charges, which amount to 350 DM/annum with the use of exhaust gas and oil firing, there is still an appreciable saving in total annual cost, which justifies in every case the provision of this type of boiler.

In Table 14, we have compared the annual heating costs of equipment with and without the use of exhaust gas, which shows the advantages of using the exhaust gas and oil fired boiler.

The example selected, with an average speed of 50 km/h, is the most unfavourable for the exhaust gas and oil fired boiler, as the low average speed means that the power developed by the motor is low and consequently so is the available quantity of exhaust gas. With a higher average speed, the annual saving in cost of heating increases considerably, as shown in figures 15 and 16, in which, at the same time, we are comparing only the cost of fuel alone because the two types of heating boiler involve the same expense apart from capital charges. We can, however, always deduct the additional 350 DM/annum charged to the exhaust gas and oil fired boiler to obtain the net saving.

TABLE 14. — **Annual cost of heating for a ten or five coach train, steam heating from oil-fired, or exhaust gas and oil-fired boiler, $t_{ü}$ = 15° C (1 965 heating hours).**

Type of heating	*Steam with oil-fired boiler*	*Steam with oil-fired and exhaust gas boiler*	*Saving*
10 coach set	30 620	23 800	6 820
5 coach set	15 100	11 520	3 580

Whilst an average of V = 50 km/h and 2 000 heating hours per annum gives with the use of exhaust gas and oil firing a saving of about 6 500 DM/annum on the total costs, this saving is increased to about 20 000 DM/annum for a ten coach set when the average speed reached 80 km/h.

Each Railway Administration can thus easily appreciate the economic advantages of using exhaust gas from the Diesel motors for heating passenger trains.

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[625 .285 (44)]

Prototype Budd railcar X. 2051 of the French National Railways (S.N.C.F.).(*)

Fig. 1.

The French National Railways at the beginning of 1957, put into service a Budd railcar, based on the Budd R.D.C. type railcars used in the U.S.A.

This railcar was built by "Etablissements CAREL, FOUCHÉ & Cie"; a general view of the car is shown in figure 1 and a diagram of the interior arrangement is shown in figure 2. It comprises :

— a driving compartment at each end;

(*) Article drafted from information supplied by the General Management of the French National Railways.

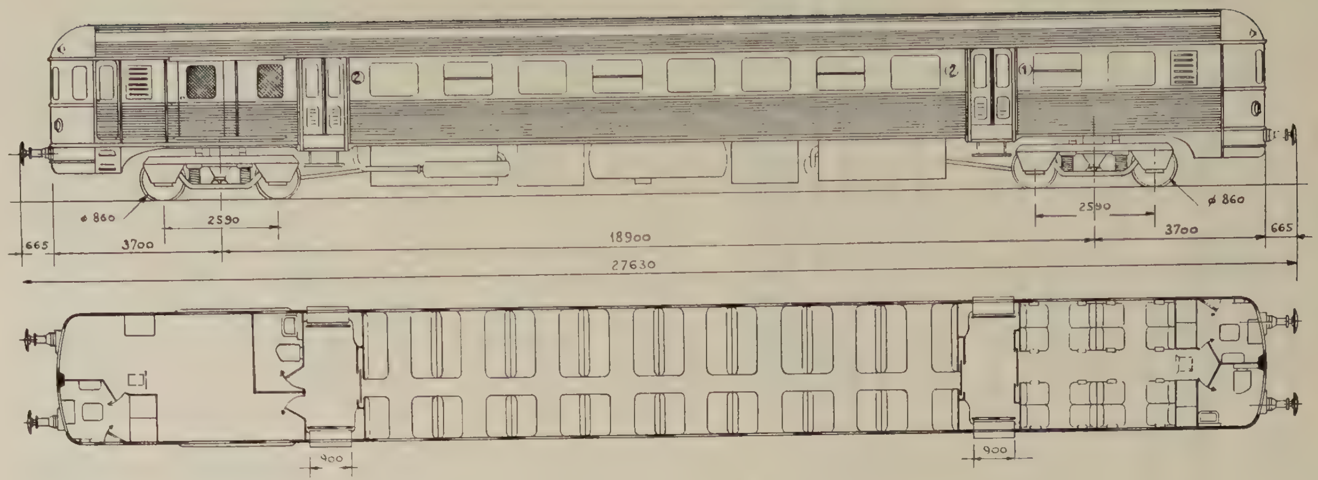

Fig. 2.

Power of the Diesel motors at 1 800 r.p.m. . . . 2 × 275 HP

Weight in running order . . . . 47,5 t

Speed limit . . . . 140 km/h (87 m.p.h.)

Hydro-mechanical transmission.

« ROLOKRON » anti-slip device.

Disc brakes.

Heating by hot-water radiators and fans.

Lighting with fluorescent tubes.

Forced ventilation.

Capacity:

Driving compartment		2
1st class seats		16
2nd class seats		80
Baggage compartment	useful area	$9\ m^2$
	load	3 300 kg
Toilets		2

— a large second-class compartment for 80 passengers;
— a small first-class compartment for 16 passengers;
— two entrance vestibules;
— a baggage compartment with an available floor space of 9 m^2;
— two toilet compartments.

The weight of the car, in working order, is 47.5 tons.

The body is stainless steel, welded in accordance with the Budd Shotwell process by CAREL, FOUCHÉ & Cie.

Air-conditioning is not provided. Only four window lights grouped on one end can be partly opened, but forced ventilation has had to be provided since the cars will run mostly with all windows closed.

Heating of the railcar is by hot-water radiators and fans.

Motor equipment comprises two horizontal Diesel engines, two-stroke, supplied by GENERAL MOTORS (U.S.A.), of 275 HP. at 1 800 r.p.m. The motors are spring-mounted under the floor and each drives one axle of one bogie through ALLISON hydro-mechanical transmission and torque converters; they provide a speed of 140 km/h (87 miles per hour).

This new railcar will allow the S.N.C.F., by comparing it with their standard railcars, to assess the advantages and disadvantages of underfloor engines and to test the efficiency of certain American techniques : motor, hydro-mechanical transmission, disc brakes, anti-slip device.

[625 .213]

An oscillating system with a single mass with dry frictional damping subjected to harmonic vibrations,[1]

by P. Van Bommel,

Engineer of the Netherlands Railways.

(*De Ingenieur*, No. 10, 8 March 1957.)

1. Introduction.[2]

The object of this calculation is to arrive at a theory of the vibrations transmitted by the permanent way to the carriage body through the spring gear and it is accepted that the damping occurring in the spring gear possesses the characteristics of a friction damper (3).

As a first approximation the vehicle is considered as an oscillating system with the mass concentrated at a point lying above the spring, the lower end of the spring being animated by a vertical harmonic movement and a dry frictional damper arranged in parallel with the spring.

To solve this problem, we started from the method of calculation given by den Hartog for an analogous system (oscillating system with concentrated mass with dry damping subjected to a harmonic force on the axis of the spring, the extremity of which not connected to the concentrated mass is fixed) (4).

We found an exact solution to the problem set us, that of the oscillating system with single mass. After studying this calculation, it will be seen to be useless to endeavour to arrive at an exact solution for a more complicated system with dry frictional damping. This is why we have established an approximate method easily handled. If for the oscillating system with a single mass, the approximate solution corresponds relatively well to the exact solution, we can admit reasonably that the more complicated oscillating solution can be treated in a simplified way with sufficient accuracy.

The approximation consists in replacing the dry frictional damper by an oil damper (damping force proportional to the speed), which in its cycle converts into heat an equivalent quantity of energy. We have not established the approximate method for the dry frictional damper alone, but have

(1) Summary of the dedication by Prof. Ir. H.C.A. van Eldik Thieme for the paper written by P. Van Bommel to obtain the diploma of mechanical engineer.

(2) Details on the calculations are not given. Further information concerning the calculations may be found in the report F 80 of the Research Dept. of the Netherlands Railways (Ir. P. Van Bommel). « An oscillating system with a single mass with dry frictional damping subjected to harmonic vibrations » (1955), 139 pages. This report may be obtained from the above mentioned department against payment.

(3) This investigation is already mentioned in the publication *De Ingenieur*, No. 27 (1956), page o.73 of the article : « Het dynamisch onderzoek van rijtuigbakken », by F. Bijl, Ir. and A.D. de Pater, Dr. Ir.

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generalised it to cover all types of dampers in which the damping force is an exponential function of the speed, so that the dry frictional damper is only a particular case.

2. **Exact method.**

case of the movement without stops.

As the direction of the frictional force W depends upon the relative speed of the movement of the mass and of the forced oscillation, the simplest method consists in starting from the relative movement. Figure 1 represents the schematic diagram of

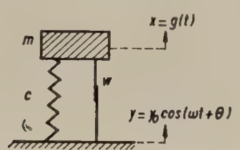


Fig. 1. — Schematic diagram of the spring gear of a vehicle.

the oscillating system whilst figure 2 gives a qualitative image of a well defined case : we will follow this figure.

The oscillation impressed on the base surface is represented by :

$$y = y_0 \cos(\omega t + \Theta) \qquad (2.1)$$

Θ is the phase angle between the relative and forced movements whilst the angle φ which will be dealt with later is the phase angle between the absolute and forced movements. The relative movement is represented by :

$$z = f(t) \qquad (2.2)$$

or for :

$$t = 0 \, , \dot{z} = 0$$

so that z attains for $t = 0$ a maximum value, this with a view to a judicious choice of the limiting values. The absolute movement is finally represented by :

$$x = g(t) \qquad (2.3)$$

so that we get the following relation :

$$z = x - y \qquad (2.4)$$

We will represent in what follows by c the spring constant, by m the mass and by F the maximum frictional force (constant).

As a preliminary we ought to make the following supposition : The movement of the mass should have the same frequency as the forced oscillation and in the half period during which the difference between the two movements grows, follows the same law as in the half period during which this difference diminishes; we will return to this later. These two suppositions are incontestibly justified because it is a question of stationary vibrations.

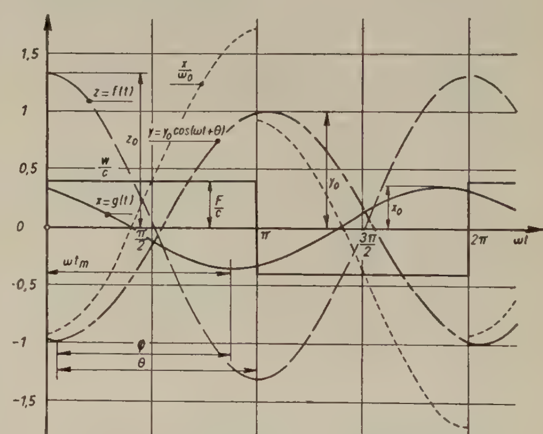


Fig. 2. — Representation of several magnitudes as a function of the time in the case in which no stop occurs.

We have selected :

$$\omega_0 = 1 \frac{\text{rad}}{\text{sec}} \, , \lambda = 2, y_0 = 1 \text{ cm}, c = 300 \frac{\text{kg}}{\text{cm}}$$

$$F = 120 \text{ kg} \, , \delta = 0.4$$

The zero of the time base is the instant at which the function z passes through its maximum positive value. The friction force changes then in sign for :

$$t = 0 \text{ and } t = \frac{\pi}{\omega}$$

from which we conclude that the curve of the movement of the mass changes curvature sharply at this instant because the acceleration undergoes a discontinuous variation. The differential equation which governs the problem is :

$$m\ddot{x} = -c(x-y) - W$$

with :

$$W = F sgn\ \dot{z} \tag{2.5}$$

During the half period :

$$0 < t < \frac{\pi}{\omega},$$

z is < 0 (see fig. 1) and the differential equation can be written :

$$m\ddot{x} + c(x-y) - F = 0 \tag{2.6}$$

From (2.1) and (2.4) and after introduction of the frequency proper to the non-damped system :

$$\omega_0 = \sqrt{\frac{c}{m}} \tag{2.7a}$$

and of the magnitude :

$$\frac{F}{c} = f \tag{2.7b}$$

we get from (2.6) the following equation :

$$\ddot{z} + \omega_0^2 (z - f) = \omega^2 y_0 \cos(\omega t + \Theta) \tag{2.8}$$

The solution of the first number, equal to zero, is

$$z = C_1 \sin \omega_0 t + C_2 \cos \omega_0 t$$

whereas the particular integral is :

$$z = \alpha y_0 \cos(\omega t + \Theta) + f$$

in which α is determined by :

$$\alpha = \frac{\lambda^2}{1-\lambda^2} \tag{2.9}$$

with :

$$\lambda = \frac{\omega}{\omega_0} \tag{2.10}$$

The general equation of the relative movement is therefore :

$$z = C_1 \sin \omega_0 t + C_2 \cos \omega_0 t + \alpha y_0 \cos(\omega t + \Theta) + f \tag{2.11}$$

Based on figure 2, it is necessary to note the following limiting values :

$$\begin{aligned} z = z_0,\ \dot{z} = 0 \quad &\text{for} \quad t = 0 \\ z = -z_0,\ \dot{z} = 0 \quad &\text{for} \quad t = \frac{\pi}{\omega} \end{aligned} \tag{2.12}$$

By means of the conditions corresponding to $t = 0$, we find the integration constants C_1 and C_2 and we get :

$$\begin{aligned} z = {} & z_0 \cos \omega_0 t + f(1 - \cos \omega_0 t) \\ & + \alpha y_0 \{ \cos \Theta (\cos \omega t - \cos \omega_0 t) \\ & + \sin \Theta (\lambda \sin \omega_0 t - \sin \omega t) \} \end{aligned} \tag{2.13}$$

whereas with the two others we can calculate z_0 and Θ.

The introduction of the factors :

$$\beta = \frac{1}{\lambda} = \frac{\omega_0}{\omega} \tag{2.14}$$

$$\zeta = \beta tg \frac{1}{2} \beta\pi \tag{2.15}$$

$$\frac{z_0}{y_0} = \tau$$

(factor of impulsion of the relative movement) (2.16)

$$\frac{f}{y_0} = \frac{F}{cy_0} = \delta$$

(factor of damping) (2.17)

gives finally :

$$\cos \Theta = \frac{\tau}{\alpha}$$

and (2.18)

$$\sin \Theta = -\frac{\delta\zeta}{\alpha}$$

The elimination of Θ from these two equations gives us an expression of τ :

$$\tau = \sqrt{\alpha^2 - (\delta\zeta)^2} \tag{2.19}$$

In the figures 3 and 4, the full lines represent respectively the characteristics of the amplitude and of the phase.

The solution given above for the relative movement is only applicable to the extent that the supposed direction of the damping

force F is exact : that is to say that during the half period :

$$0 < t < \frac{\pi}{\omega}$$

the derivative of the function z ought to remain negative. Let us then differentiate (2.13) and pose as condition that :

$$\dot{z} < 0$$

for :

$$0 < t < \frac{\pi}{\omega}.$$

We then find on taking into account (2.15) :

$$\frac{\tau}{\delta} > \frac{\beta \sin \omega_0 t + \zeta (\cos \omega t - \cos \omega_0 t)}{\sin \omega t}$$

$$\text{for } 0 < t < \frac{\pi}{\omega} \qquad (2.20)$$

For $t = 0$, the value of the second term is equal to β^2.

By numerical calculation, we must prove that the maximum of the second term for $\beta < 2$ (therefore $\lambda > 0.5$) is equal to β^2. Consequently, it requires :

$$\frac{\tau}{\delta} > \beta^2$$

during the half period :

$$0 < t < \frac{\pi}{\omega}$$

and for $\lambda > 0.5$.

The elimination of δ by means of (2.19) gives an expression of the limiting value of τ as a function of λ below which our deduction is no longer allowable :

$$\tau_g = \sqrt{\frac{\alpha^2}{1 + (\lambda^2 \zeta)^2}} \qquad (2.21)$$

In figures 3 and 4, the limiting lines are shown so that the full lines giving the frequency characteristics end at these limiting lines. The limiting lines have, in the zone of the frequencies $\lambda < 0.5$ a curious rhythm because ζ oscillates rapidly between $-\infty$ and $+\infty$ and the maximum of (2.20) is then difficult to determine. A determination carried much further of this limiting line is left aside because it would involve a special research which would take too much time. Moreover,

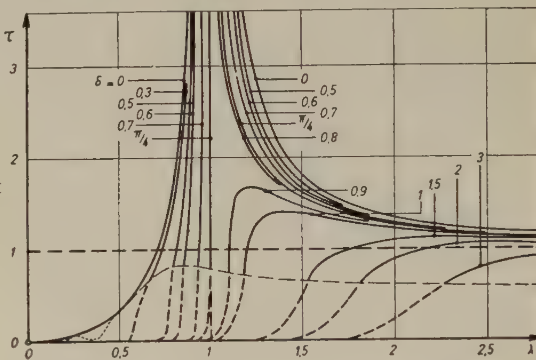


Fig. 3. — Characteristic amplitude — frequency of the relative movement.

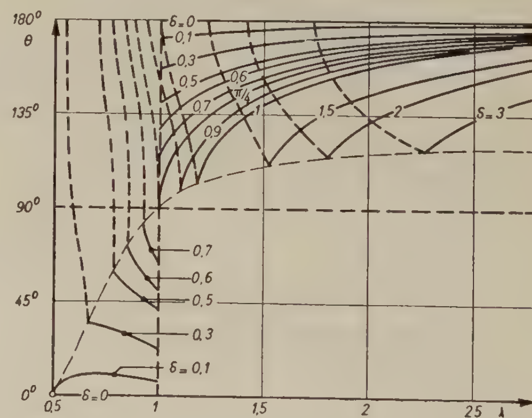


Fig. 4. — Characteristic phase — frequency of the relative movement.

this zone is of no importance because the vibrations of the vehicles have never or rarely so low a frequency and in addition we enter a zone (except for slight damping) in which there is no longer absolutely any relative movement. Seizures then occur and these will be dealt with later. We

can however point out that the limiting line touches the line $\delta = 0$ for :

$$\lambda = \frac{1}{2}, \frac{1}{4}, \frac{1}{6}, \quad \text{etc.,}$$

so that the rhythm has to be given in full; the same problem also presents itself in DEN HARTOG's calculation.

We can, now the relative movement is completely analysed, pass to the determination of the absolute movement of the mass and draw down the frequency characteristics. We have in fact from (2.4) :

$$x = z + y$$

We start from (2.13) expression of the relative movement after the two integration constants have been calculated and superpose there on the forced oscillation of (2.1). We then get for x the following expression :

$$\frac{x}{y_0} = \tau \cos \omega_0 t + \delta - \delta \cos \omega_0 t + \alpha \cos \Theta \left\{ \cos \omega t - \cos \omega_0 t + \frac{\cos \omega t}{\alpha} \right\} + \alpha \sin \Theta \left\{ \lambda \sin \omega_0 t - \sin \omega t - \frac{\sin \omega t}{\alpha} \right\} \qquad (2.22)$$

Let us eliminate τ by means of (2.18) and introduce the magnitudes α and ζ (see (2.9) and (2.15); we finally obtain the expression of x as a function of the time :

$$\frac{x}{y_0 \delta} = 1 - \frac{\cos \beta(\omega t - \frac{\pi}{2})}{\cos \frac{1}{2} \beta\pi} + \frac{\cos(\omega t + \Theta)}{\delta(1 - \lambda^2)} \qquad (2.23)$$

This is allowable so far as what preceeds is also allowable.

To find the optimum value of the movement, that which interests us most, we have to find the maximum of (2.23).

The maximum occurs for $t = t_m$ and t_m can be found by solving the transcendent equation resulting from the differentiation of (2.23) :

$$\frac{\lambda}{\delta (1 - \lambda^2)} \sin (\omega t_m + \Theta) = \frac{1}{\cos \frac{1}{2} \beta\pi} \sin \beta (\omega t_m - \frac{\pi}{2}) \qquad (2.24)$$

For a given value of the damping and of the frequency, we can calculate ωt_m and by substitution in (2.23) we find the amplitude of the movement of the mass. Let us call this amplitude x_0 when we can define the factor of impulse σ (without dimension) :

$$\sigma = \frac{x_0}{y_0} \qquad (2.25)$$

The phase angle φ (angle between the absolute movement and forced movement), can be found starting from figure 1 by the following formula :

$$\varphi = \Theta + \omega t_m - 180^o \qquad (2.26)$$

The figures 5 and 6 represent the two frequency characteristics : we only consider provisionally the full lines for which our deduction is allowable in the same sense as has been said about the relative movement. The limiting lines are also transferred to the diagrams by very careful calculation of the frequencies for which for a given damping, the limiting value τ_g is given, which is accompanied by a great amount of numerical work of great accuracy.

Let us again consider figure 2; we see that the various functions are equally drawn for the half period $\pi < \omega t < 2\pi$. The numerical values of the functions for determined instants are equal in absolute value but of another sign, or in formula :

$$f(t) = -f\left(t + \frac{\pi}{\omega}\right) \qquad (2.27a)$$

$$g(t) = -g\left(t + \frac{\pi}{\omega}\right) \qquad (2.27b)$$

It is after all easy to see, if we displace the zero of the time base a distance π, replace F by —F and the limit values (2.12) by :

$$z = -z_0 \quad , \dot{z} = 0 \quad \text{for} \quad t = 0$$

$$z = z_0 \quad , \dot{z} = 0 \quad \text{for} \quad t = \frac{\pi}{\omega}$$

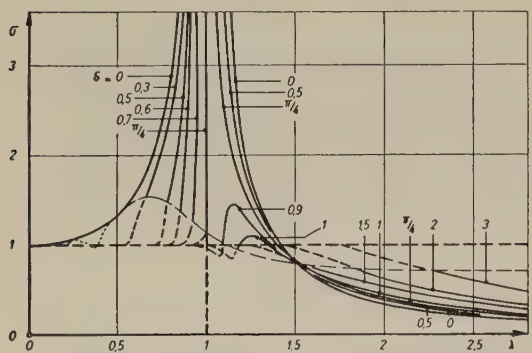


Fig. 5. — Characteristic amplitude — frequency of the absolute movement.

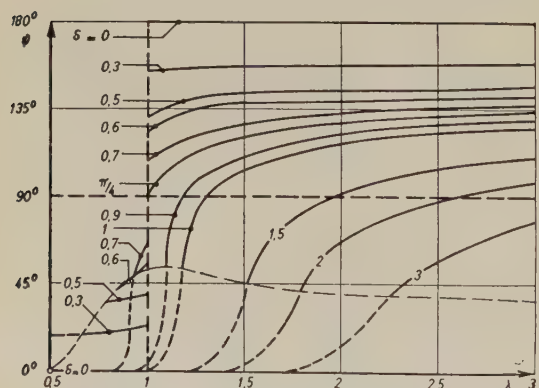


Fig. 6. — Phase — frequency characteristic of the absolute movement.

The forced ventilation is, in addition represented by :

$$y = -y_0 \cos(\omega t + \Theta)$$

In (2.13) all we have to do is to replace z_0, f and y_0 respectively by $-z_0$, $-f$ and $-y_0$, so that (2.27a) is demonstrated. In the same way too (2.27b) can be.

We have seen so far that the deduction made had only a restricted value. The relative speed should remain negative during the half period :

$$0 < t < \frac{\pi}{\omega};$$

in other terms, the relative movement has, during this half period, a displacement which retains the same direction. Consequently, there is the possibility that the relative speed at a certain moment (compared in the half period) may be equal to zero and that there may be a stop in the relative movement. We will now proceed to a more detailed examination of this problem of the stops.

3. Exact method.

Case of a stop per half period.

Figure 7 gives a qualitative image of a given case and we will take this figure as the starting point of our calculation. A stop occurs here in the relative movement. The zero of the time base is so chosen that the relative movement is in the positive direction at the end of a stop.

The differential equation (2.6) is applicable to the case $z \neq 0$. Its resolution is carried up to and including (2.11) in an analogous way to that we have seen. The essential difference shows itself when establishing the limiting values.

It is now necessary to introduce like 5th unknown, the instant t_0 giving the moment at which a stop begins in the negative direction. Consequently, a relative movement is possible during the period : $0 < t < t_0$ with a stop during :

$$t_0 < t < \frac{\pi}{\omega}.$$

The limiting values are now :

$$\begin{aligned} z &= z_0 \quad , \dot{z} = 0 \quad \text{for} \quad t = 0 \\ z &= -z_0 \, , \dot{z} = 0 \quad \text{for} \quad t = t_0 \end{aligned} \tag{3.1}$$

The 5th equation required for finding the unknown 5th can be found by considering what passes just before the instant $t = 0$: at this moment a result :

$$\ddot{z} = 0 .$$

This is also valid for the moment $t = \varepsilon$, the frictional force being then nearly F. Introducing these data in the differential equation (2.6) when ε tends towards zero :

$$- cz_0 + F + \lambda^2 y_0 \cos \Theta = 0.$$

The introduction of these values without dimension gives :

$$\tau - \delta = \lambda^2 \cos \Theta \tag{3.2}$$

Using the 2 first limiting values of (3.1), we can determine the integration constants C_1 and C_2 : in this way we obtain the expression (2.13). By using the two other limiting values and (3.2), we obtain after much calculation and the introduction of dimensionless magnitudes, the following equations :

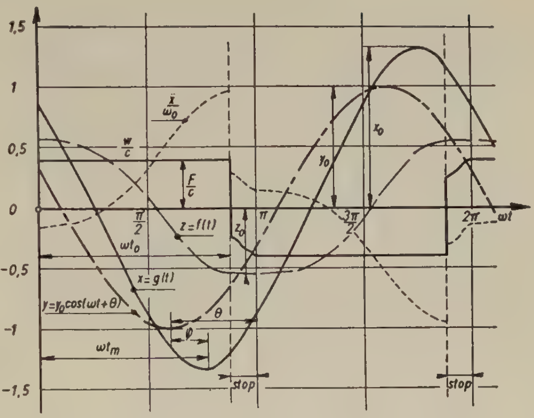


Fig. 7. — Representation of various magnitudes as functions of the time in the case in which there is one stop per half period.

We have taken :

$$\omega_0 = 1 \frac{\text{rad}}{\text{sec}}, \lambda = 0.69, y_0 = 1 \text{ cm}, c = 300 \frac{\text{kg}}{\text{cm}}$$

$$F = 120 \text{ kg} , \delta = 0.4$$

$$\cos \Theta = \pm \frac{\cos \beta\omega t_0 - \cos \omega t_0}{\sqrt{(\cos \beta\omega t_0 - \cos \omega t_0)^2 + (\lambda \sin \beta\omega t_0 - \sin \omega t_0)^2}} \tag{3.3a}$$

$$\sin \Theta = \pm \frac{\sin \omega t_0 - \lambda \sin \beta\omega t_0}{\sqrt{(\cos \beta\omega t_0 - \cos \omega t_0)^2 + (\lambda \sin \beta\omega t_0 - \sin \omega t_0)^2}} \tag{3.3b}$$

$$\frac{2\delta}{\alpha} = \pm \frac{(\cos \omega t_0 - \cos \beta\omega t_0)(1 - \lambda^2 + \cos \omega t_0 - \lambda^2 \cos \beta\omega t_0) + (\sin \omega t_0 - \lambda \sin \beta\omega t_0)^2}{\sqrt{(\cos \beta\omega t_0 - \cos \omega t_0)^2 + (\lambda \sin \beta\omega t_0 - \sin \omega t_0)^2}} \tag{3.4}$$

Since the numerator of (3.4) is always positive for :

$$0 < t_0 < \frac{\pi}{\omega} \text{ and } \beta < 2 ,$$

in these equations the sign + must be used for $\lambda < 1$ and the sign — for $\lambda > 1$. This results from the fact that δ is positive in nature and α becomes negative for $\lambda > 1$. Consequently, it is not possible to find explicit expressions for τ, Θ and t_0 so that they have to be found by numerical solution of the three transcendant equations (3.3a), (3.4) and 3.2). For this, we determine a certain value of λ (therefore β), the magnitudes δ, τ and Θ as function of ωt_0 by substitution of several values of ωt_0 in the three equations mentioned above.

It is important to point out that the phase angle Θ (as well as φ) is defined as being the out of phase angle behind the maximum of the relative movement (resp. absolute) relatively to the maximum of the forced oscillation. This contrarily to the viscous damping, where a definition of the phase angle is directly at hand when the forced oscillation is purely harmonic.

Being linked with the sign given at the beginning to the frictional force, we can only adopt for ωt_0 the values comprised between 0 and 180°.

It is clear that when $\omega t_0 = 0$ absolutely no further relative movement is produced and the system may be said to be blocked. By substituting $\omega t_0 = 0$ in (3.4), we find the values of damping for which the system is blocked for a given frequency. By development in series of the right hand member of (3.4), this becomes for $\omega t_0 = 0$:

$$\frac{2\delta}{\alpha} = 2(1 - \lambda^2)$$

From this we obtain :

$$\delta = \lambda^2 \tag{3.5}$$

By means of l'HOPITAL's theorem, we can determine the one or the other thing for $\lambda = 1$. We can find in this way that the limiting line of the figure 3 has the value :

$$\frac{\pi}{4} \text{ for } \lambda = 1$$

this being linked with the fact that for $\omega t_0 = 180°$, there has so far been no stop which consequently is linked with the determination of the limiting line.

After the above, we can envisage the absolute movement of the mass in the case in which a stop occurs in the relative movement. We start from (2.13) after the two integration constants have been determined. We have in addition $x = z + y$ so that after several simplifications we obtain the following equation :

$$\begin{aligned}\frac{x}{y_0} &= (1 + \alpha) \cos (\omega t + \Theta) \\ &+ (\tau - \delta - \alpha \cos \Theta) \cos \omega_0 t \\ &+ (\alpha\lambda \sin \Theta) \sin \omega_0 t + \delta \end{aligned} \tag{3.6}$$

This equation represents the displacement of the mass as a function of the time. It is only valid for $0 < t < t_0$. Taking

$$t_0 < t < \frac{\pi}{\omega},$$

the differential equation is really valid, but the frictional force W which is found therein is not constant during this period. To determine x making :

$$t_0 < t < \frac{\pi}{\omega},$$

we can make use of the equation :

$$\frac{x}{y_0} = \tau + \cos (\omega t + \Theta) \tag{3.7}$$

since the magnitude of z during this period is z_0 (or τy_0).

To determine the maximum value x_0 of x, we have to calculate the maximum value of (3.6) and (3.7). The maximum occurs when $t = t_m$. We suppose temporarily that the maximum value shows itself for an instant between 0 and t_0. For this it is not enough for $t_m < t_0$ because (3.6) is only valid by making $0 < t < t_0$ and it is consequently perfectly possible making :

$$t_0 < t < \frac{\pi}{\omega}$$

there may be a value of x which may be greater than that which would be given by (3.6). After doing so, it is always necessary to check it and if need be correct it. Differentiation of (3.6) gives an equation from which we can drawn t_m :

$$\begin{aligned}\lambda (1 + \alpha) \sin (\omega t_m + \Theta) \\ = - (\tau - \delta - \alpha \cos \Theta) \sin \beta\omega t_m \\ + (\alpha\lambda \sin \Theta) \cos \beta\omega t_m \end{aligned} \tag{3.8}$$

The phase angle φ is found by :

$$\varphi = \Theta + \omega t_m - 180° \tag{3.9}$$

We can now design the frequency characteristics for the case in which the stop occurs (broken lines in figs. 5 and 6). Therein are represented the phenomenon of the system which is blocked in a certain frequency zone for each damping.

Finally the rhythm of the frictional

force in the period during which the relative movements undergoes a stop :

$$\left(t_0 < t < \frac{\pi}{\omega}\right)$$

can be found by the equation :

$$\frac{W}{cy_0} = \delta_w = \tau - \lambda^2 \cos(\omega t + \Theta) \qquad (3.10)$$

since in (2.8) can be substituted :

$$\ddot{z} = 0 \quad \text{and} \quad z = z_0$$

4. Exact method. The acceleration.

Naturally, it is possible to determine the acceleration starting from the preceding equations. However, we may calculate quickly the maximum accelerations by means of the differential equation first used, as well for the case without stop as for the case with stops in the relative movement. The differential equation is valid in the two cases. It can also be written :

$$m\ddot{x} = F - c(x - y)$$

or

$$\frac{\ddot{x}}{y_0\omega^2} = \delta - \frac{z}{y_0}$$

The maximum value of the acceleration shows itself consequently when $(-z)$ is maximum which is the case for $t = \pi$ (without stops) respectively :

$$t_0 < t < \frac{\pi}{\omega}$$

(a stop) for z it is necessary to take $-z_0$. Let us define the dimensionless impulse factor σ_v as being :

$$\sigma_v = \frac{\ddot{x}_{max}}{y_0\omega_0^2} \qquad (4.1)$$

We then find :

$$\sigma_v = \delta + \tau \qquad (4.2)$$

Figure 8 gives the corresponding frequency characteristics.

In case of a stop, the amplitude curves of the accelerations running the length of the parabola : $\sigma_v = \lambda^2$.

The fact that the accelerations for example for : $\delta = 0.3$ in the infra-critical zone are superior to the accelerations for $\delta = 0$ has been capable of being verified subsequently.

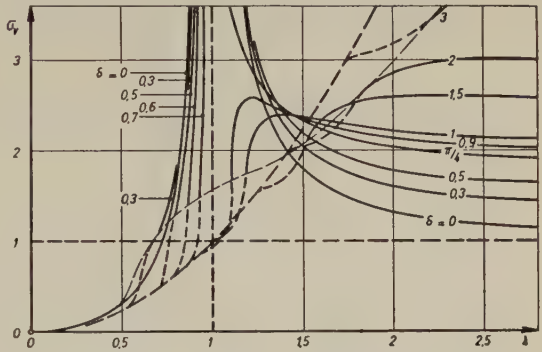


Fig. 8. — Characteristic of the acceleration amplitude - frequency of the absolute movement.

5. Approximative method.

Before dealing with the approximate solution of the oscillating system with single mass and dry friction damping, we give first of all a general approximation for all sorts of damping in which the frictional force is an exponential function of the speed.

We can represent, in a general way, the damping force by :

$$K = k_n \, |\dot{x}|^n \, sgn \, \dot{x} \qquad (5.1)$$

To simplify it, let us replace this damping force by the force :

$$K' = k_1 \dot{x} \qquad (5.2)$$

supposing that this latter transforms into heat, by cycle, an equivalent quantity of energy as the primitive damper.

The energy E_1 which is transformed by cycle in heat by the replacement damper is :

$$E_1 = \int_0^{\frac{2\pi}{\omega}} k_1 \dot{x} dx = \pi k_1 \omega x_0^2 \qquad (5.3)$$

The energy E_n transformed in heat by cycle by the primitive damping is :

$$E_n = 4\int_0^{\frac{\pi}{2\omega}} k_n \dot{x}^n dx = 4\int_0^{\frac{\pi}{2\omega}} k_n \dot{x}^{n+1} dt \quad (5.4)$$

These two energies ought to be equal, which with $x = x_0 \sin \omega t$, gives :

$$k_n x_0^{n-1} \omega^{n-1} \int_0^{\frac{\pi}{2}} \cos^{n+1} \omega t d(\omega t) = \frac{\pi}{4} k_1$$

From the theory of gamma functions, we know :

$$\int_0^{\frac{\pi}{2}} \cos^{n+1} \omega t d(\omega t) = \frac{1}{2}\sqrt{\pi} \frac{\Gamma\left(\frac{n+2}{2}\right)}{\Gamma\left(\frac{n+3}{2}\right)}$$

To be complete let us note :

$$\Gamma(1) = \Gamma(2) = 1 \; ; \; \Gamma\left(\frac{1}{2}\right) = \sqrt{\pi}$$

$$\Gamma(n+1) = n! = n\Gamma(n) \; ;$$

$$\Gamma\left(\frac{n}{2}\right) = \frac{n-2}{2} \cdot \frac{n-4}{2} \cdots \frac{1}{2}\sqrt{\pi}$$

Let us pose in addition :

$$\frac{2}{\sqrt{\pi}} \frac{\Gamma\left(\frac{n+2}{2}\right)}{\Gamma\left(\frac{n+3}{2}\right)} = \gamma_n \quad (5.5)$$

We can then write the constant of proportionality k_1 as follows :

$$k_1 = k_n \gamma_n x_0^{n-1} \omega^{n-1} \quad (5.6)$$

As it is now a question of viscuous damping, the differential equation can be written :

$$m\ddot{x} + k_1 \dot{x} + cx = cy_0 \sin \omega t + k_1 \omega y_0 \cos \omega t \quad (5.7)$$

supposing that the movement of the base surface be $y = y_0 \sin \omega t$. The expressions of the impulse factor σ and the phase angle φ are supposed known for a differential equation, so that we can obtain with (5.6) :

$$\sigma = \sqrt{\frac{1 + k^2_n \gamma^2_n x_0^{2n-2} \omega^{2n-2} c^{-2}}{(1-\lambda^2)^2 + k^2_n \gamma^2_n x_0^{2n-2} \omega^{2n-2} c^{-2}}} \quad (5.8)$$

$$tg\varphi = \frac{m\omega^3 k_n \gamma_n x_0^{n-1} \omega^{n-1}}{c^2(1-\lambda^2) + \omega^2 k^2_n \gamma^2_n x_0^{2n-2} \omega^{2n-2}} \quad (5.9)$$

Let us introduce the following function :

$$f_n = \frac{k_n \gamma_n \omega_0{}^n y_0{}^{n-1}}{c} \quad (5.10)$$

We so find for (5.8) the following expression :

$$\sigma = \sqrt{\frac{1 + f^2_n \lambda^{2n} \sigma^{2n-2}}{(1-\lambda^2)^2 + f^2_n \lambda^{2n} \sigma^{2n-2}}} \quad (5.11)$$

which leads to the equation from which can be drawn σ for each particular case :

$$\sigma^2(1-\lambda^2) + f^2_n \lambda^{2n} \sigma^{2n} = 1 + f^2_n \lambda^{2n} \sigma^{2n-2} \quad (5.12)$$

The expression of the phase angle φ becomes after several transformations :

$$tg\varphi = \frac{f^n \lambda^{2n+2} \sigma^{n-1}}{(1-\lambda^2) + f^2_n \lambda^{2n} \sigma^{2n-2}} \quad (5.13)$$

Let us also suppose known for an oscillation problem controlled by a differential equation of the type (5.7) the condition of free frequency to be :

$$4\,c\,m > k^2_1 \quad (5.14)$$

which after introducing the function f_n becomes :

$$2 > f_n \sigma^{n-1} \lambda^{n-1} \quad (5.15)$$

Based on the general deduction given above, it becomes easy to determine the desired magnitude in the various cases. Let us illustrate that for $n = 0$. We can also make k_n equal to the frictional force F.

The magnitude λ_n defined by (5.5) becomes

$$\frac{4}{\pi}$$

so that we find for the function f_n :

$$f_0 = \frac{4}{\pi} \frac{F}{cy_0} \qquad (5.16)$$

Starting from (5.12), we find after substitution an equation of the second degree for σ^2 :

$$\sigma^4(1 - \lambda^2)^2 + \sigma^2(f^2{}_0 - 1) - f^2{}_0 = 0. \qquad (5.17)$$

From this we draw the impulse factor σ and find after having interpreted the sign, the following expression :

$$\sigma = \frac{1}{|1 - \lambda^2|}\sqrt{\frac{1 - f^2{}_0}{2} + \sqrt{\left(\frac{f^2{}_0 - 1}{2}\right)^2 + f^2{}_0(1 - \lambda^2)^2}} \qquad (5.18)$$

Starting from (5.9), we then find the phase angle φ :

$$tg\varphi = \frac{f_0\lambda^2|\sigma|}{f^2{}_0 + (1 - \gamma^2)\sigma^2} \qquad (5.19)$$

For greater ease we retain in the formulae the parameter f_0 instead of δ for the damping, which is defined by (2.17). There is the following relation between these two :

$$f_0 = \frac{4}{\pi}\,\delta \qquad (5.20)$$

By means of (5.15) and by substituting $n = 0$, we can note the condition of free periodicity. It can be written :

$$\lambda\sigma > \frac{f_0}{2} \qquad (5.21)$$

From (5.17), after some calculations, we can eliminate f_0 and retain the following formula for the limiting line of the diagram $\sigma - \lambda$:

$$\sigma_g = \frac{\sqrt{1 + 4\lambda^2}}{1 + \lambda^2} \qquad (5.22)$$

Thus, through the approximation, we fall back on the linear problem and as the movement of the mass is represented by a harmonic movement of frequency ω, we can write directly the expression of the dimensionless amplitude of the acceleration :

$$\sigma_v = \frac{x_0\omega^2}{y_0\omega^2{}_0} = \lambda^2\sigma \qquad (5.23)$$

Figures 9, 10 and 11 give the corresponding frequency characteristics as well as the limiting line of free periodicity.

Some observations and conclusions.

1. In the oscillation problem dealt with by DEN HARTOG (same mass oscillation system but with a harmonic side acting on the mass), it is also possible that per half period there may be more than one

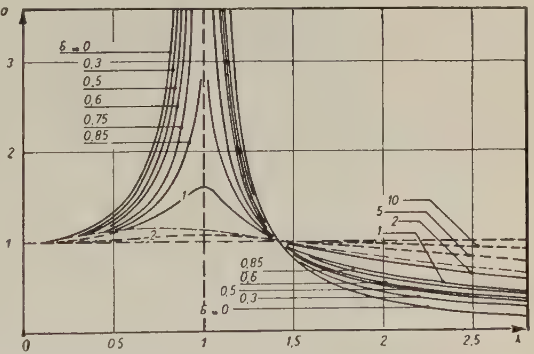


Fig. 9. — Amplitude - frequency characteristic (approximate solution).

stop. In our case, this does not occur because there is a stop during $0 < t < t_0$ the relative speed should be $\dot{z} < 0$. Furthermore during :

$$t_0 < t < \frac{\pi}{\omega}$$

the following inequality should be valid :

$$|my_0\omega^2 \cos(\omega t + \Theta) - cx_0| < F$$

or even after division by cy_0 :

$$|\lambda^2 \cos(\omega t + \Theta) - \tau| < \delta$$

Seeing that this condition has always

been met in the dampings we have considered, so long as there were no blocking, we can almost certainly suppose there will be no case in which more than one stop in a half period will occur.

2. When the acceleration characteristics are shown, it is usual to show as ordinates the following dimensionless factor :

$$\sigma_v = \frac{x_0 \ \omega^2}{y_0 \omega^2{}_0}$$

In fact, when we know y_0 and ω_0, this method of expression has indeed a meaning. However, to better appreciate the quality of the spring gear as regards the accelerations, we can take the ratio between the acceleration which the elastically sprung mass would undergo and that when the mass was not sprung. We obtain in this way the dimensionless factor.

$$\sigma'_v = \frac{x_0 \ \omega^2}{y_0 \ \omega^2} = \sigma$$

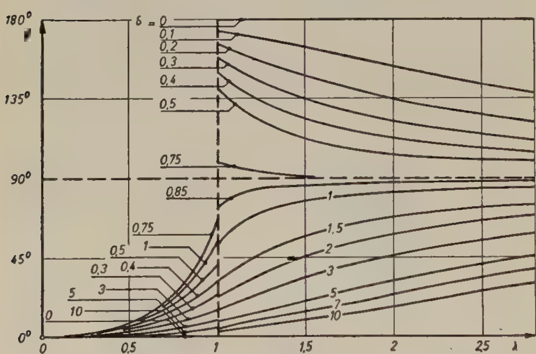


Fig. 10. — Phase - frequency characteristic (approximate solution).

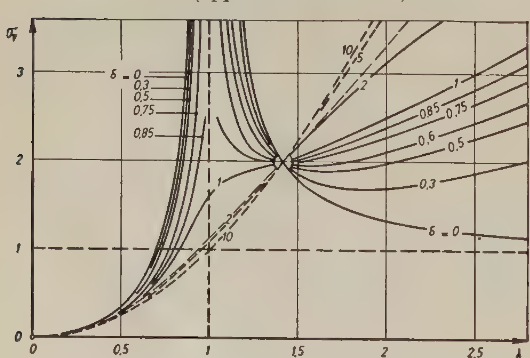


Fig. 11. — Characteristic of the acceleration amplitude - frequency (approximate solution).

3. When $\delta \leqslant \frac{\pi}{4}$, there is always, with a friction damper, an infinite impulse when the system is attacked in the frequency proper to the non damped system. This is the reverse of the viscous damper in which there is never an infinite impulse. This moreover is easy to interpret when it is remembered that the energy transformed into heat by the dry damper increases proportionally with the amplitude whereas it increases quadratically with a viscous damper.

4. The approximate solution agrees closely enough with the exact solution for the supercritical oscillations. The concordance is less close for infra-critical oscillations, the reason being found in the fact that the approximate method gives no information on the subject of a block of the system.

5. We see that in a system with frictional damping the acceleration for the high frequencies do not become great finally to become infinite. The factor σ_v approaches for the high frequencies to the limiting value of finite value $1 + \delta$. We observe furthermore from the graphical representation that the approximate method gives a different image but the concordance is relatively good in the zone of the frequencies occurring most often in the oscillation of the vehicle.

6. We cannot draw from the calculations made too many conclusions on the subject of the suspension of a vehicle in general. The suspension of a vehicle is more complicated because it is a question of oscillating systems with several masses. Then too it also offers still other phenonema.

[656 .211 (42)]

Railway ticket office equipment.

Shown at Modern Railway Travel Exhibition.

Portable unit for use on trains.

(From *The Modern Transport*, July 20, 1957.)

One of the most interesting features of the Modern Railway Travel Exhibition staged by the British Transport Commission was a booking office layout as designed by the Eastern Region, replete with examples of equipment. It has been accepted that the features of the ideal ticket office are the provision of equipment for the promotion of selling and accounting efficiency within the office, neat and attractive appearance, light but durable construction, and good lighting and economical heating. As compared with the traditional type of ticket office with its solid front and occasional small windows for transactions with the public, modern ticket offices in the Eastern Region of British Railways (and elsewhere) are being constructed with a glazed front through which can be seen almost every part of the office interior. The exhibit formed an indication of what is being carried out under the railway modernisation plan to give a speedier ticket issuing service to the public and to increase the efficiency of the working, whilst at the same time reducing overhead costs.

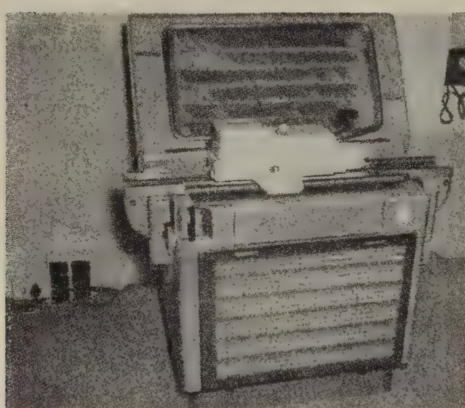
The new booking office at Wimbledon, showing Ultimatic machines either side of the guichet with Bellmatic scopes for other tickets.

The Eastern Region has recently com-

pleted the development of a wide range of steel equipment to meet the needs of the many and varied types of traffic offices at stations. Prior to this development, every office was designed individually (providing for timber construction of internal equipment), and it was necessary to draw up detailed plans each time an office was re-equipped or constructed.

Steel units.

Steel equipment on the unit principle has the advantages of easy mobility, improved appearance, longer life and easier maintenance. The range is wide enough to allow each office to be equipped to its precise requirements for efficient operation and can easily be added to or reduced to meet any changes in traffic flows or requirements. The equipment, which is based on a module of 2 ft., consists basically of a series of caseworks into which a wide variety of inserts can be fitted. In ticket offices it provides for the installation of ticket issuing machines or ticket dispensers together with all the facilities for storage of cash, ticket stocks, stationery and all other relevant materials. In parcels offices an improved type of roller counter has been developed together with facilities for weighing machines and label storage, etc.



The Westinghouse Garrard Multiprinter.

The steel equipment was developed by the Eastern Region architect's office in collaboration with a committee of departmental representatives and the manufacturers, Roneo, Limited. The layout which was exhibited displayed only a small part of the range of equipment and was based on the requirements of a ticket office fitted with the latest types of ticket issuing equipment. The first fully mechanised ticket office, equipped with steel furniture, is located at Southend-on-Sea Central, to which point some of the machines shown were transferred immediately after the exhibition.

Multiprinter.

The Multiprinter ticket printing, issuing and accounting machine is designed to facilitate the issue of frequent, but not intensive, bookings of many different types to widespread destinations. In addition to printing, numbering and dating the tickets as required, the machine records details of all issues in chronological order, including the number of issues from each individual series. It also maintains an aggregate debit which can be ascertained at any given time. It is supplied by Westinghouse Garrard Ticket Machines, Limited, Chippenham, in two sizes, the Major model and the Minor. They are capable of printing and issuing 1 260 and 630 different kinds of tickets respectively. Both machines are 4 ft. 3 in. high and 1 ft. 8 in. in depth. The Major model is 5 ft. wide and the Minor 3 ft. 6 in. Blank tickets used for feeding the machines are of standard size.

The printing plates are stored in a main cabinet, fronted by a clear glass panel. Through this panel can be seen the statistical counters, one for each plate, which advance by one unit each time the relevant print plate is brought into operation. On top of the cabinet is a movable printing carriage, and at the rear is an inclined index panel on which destination indicators are arranged in echelon form. The panel is illuminated section by section as the printing carriage is moved. To print a ticket, the operator moves the carriage until

an indicator is over the desired destination. A blank ticket of the appropriate colour is then inserted into a slot. The machine instantly prints and ejects the ticket and makes all necessary records of the issue.

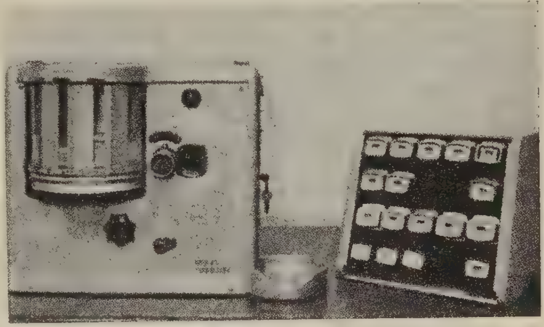
Mechanised booking office accounting.

A printed record of the state of the dissector counters can be made at any time by a special device running on guide rails. This impresses on a paper strip the identification number of each printing plate and the number of times each plate has been used. All necessary statistics can be compiled from this printed record which, in effect, mechanises the old laborious method of assessing the issues from each individual series by manual calculation. The totaliser debit can be ascertained at any time by placing a special card strip in a slot at the left of the printing carriage and depressing a handle. In practice, the ticket office clerk commences his turn of duty by printing a special ticket which shows the date, station name and machine number. On this he inserts his name and turn of duty. He then places a strip of card in the machine and ascertains the existing debit. The same procedure is observed at the termination of his turn of duty, and he is thus responsible for the difference in cash between the two debit figures.

The traditional system of accountancy to make up the ticket clerks' debit involves the manual recording of opening and closing numbers of every series of tickets in use and working out, item by item, the debit for all issues to arrive at a total debit. This involves many man-hours of labour every day according to the size of the station. The machines will produce the total debit in a few minutes. The first installation of these machines on British Railways has recently been carried out at Doncaster and other installations will follow in the near future.

Flexiprinter.

The Flexiprinter machine (also by Westinghouse Garrard) is designed primarily for the printing and issuing of tickets at small- and medium-sized ticket offices. As its name implies, however, it is flexible in its application, and is capable of issuing either ordinary tickets or season tickets. It is therefore also eminently suitable for use as a complementary machine to the Multiprinter at busy suburban stations where



The Westinghouse Garrard Flexiprinter.

the season ticket issues are very heavy. Like the Multiprinter the machine records details of all issues in chronological order and maintains an aggregate debit which can be ascertained at any given time. The fundamental difference between the two machines, however, is that, whilst the Multiprinter print plates are self-contained, the Flexiprinter is equipped with external print plates which are stored in specially designed racks adjacent to the machine.

This method makes possible a very much smaller machine, and the Flexiprinter (19 1/2 in. by 14 in. by 17 in.) is in fact approximately the size of a cash register. Basically, the machine is a small printer which is placed on a suitable stand adjacent to the rack of external printing units. It consists of a cabinet enclosing a card magazine, date roller, totalisor, record strip and mechanism. The card magazine has vertical storage columns, and may consist

of either seven ordinary ticket columns, three ordinary and three season ticket columns, or two ordinary and four season ticket columns, according to the individual requirements of the station. The external printing units consist of a hand grip internal cylinder and outer casing. In the front of each hand grip is an indication card and a statistical counter which records the number of issues from that particular print block.

Operating method.

To print a ticket, the operator selects the appropriate external printing unit, revolves the card magazine into position according to the type of card required, and inserts the print unit into the cabinet. The unit, when given a slight twist in a clockwise direction, immediately issues a completely printed, numbered and dated ticket. When the unit is held in the operative position, tickets are continuously printed and issued at the rate of 100 per min. Simultaneously, the code number of the print unit, the date, and the progressive ticket number are recorded on a detail strip, and the amount of the fare is added to a cash totalisor.

The ticket office clerk inserts a special card in the machine at the beginning and end of each turn of duty, and a simple subtraction of the two amounts gives the cash balance figures for which he is responsible. One of these machines will shortly be in service at Southend-on-Sea Central, Eastern Region.

Experiments are shortly to be conducted at Chippenham (Wiltshire) and Caerphilly (Glamorganshire) and other selected stations on the Western Region with the mechanisation of ticket issue. As long ago as 1911, the former Great Western Railway pioneered tests in this country of machines capable of printing and issuing tickets and at the same time producing an audit roll. Two appliances were installed at the newly rebuilt Snow Hill Station, Birmingham, but the experiment, while partially successful, was not pursued at the time as the passenger fares structure and booking arrangements did not readily lend themselves to mechanisation with the earlier types of machine. For some time many of the Continental railway undertakings have been using machines which have been developed from those tested at Birmingham and now the latest Major Multiprinter is being installed at Chippenham while Caerphilly will have a Flexiprinter in September.

Rapidprinter.

Also shown was the Westinghouse Garrard Rapidprinter, a heavy-duty machine designed to facilitate intensive bookings to a limited number of destinations. As such, it is capable of a speedy and sustained performance. Electrically operated, it consists of a cabinet enclosing a minimum of 10 and a maximum of 25 printing units. Upon pressure of an operating button, a blank card is fed from a reel through the appropriate printing block and on to a dater. It is then guillotined on to a delivery belt which conveys the ticket through an ejection slot to the ticket window. The operating keyboard is usually fixed flush in the booking counter at the operator's left hand, and consists of a number of push buttons equal to the number of printing units the machine is designed to accommodate. In actual operation, only a light touch is necessary for one ticket to be issued, but if pre-printing is necessary the continuous depression of a button will ensure delivery at the rate of 240 per min. An experienced operator can estimate exactly the depression required for any specific number of tickets.

As each print block is independent, it can be freely interchanged with others to meet changing demands at varying times of the day. In this way it is possible for a print block to be inserted responding to push button No. 1 that will issue early morning return tickets up to 7.30 a.m. and then be interchanged with a different block issuing ordinary return tickets. Still later in the day, a third print block issuing cheap day or excursion tickets can be inserted in place of the ordinary return block. Each

print block has on top a visible counter showing the number of tickets that have been printed through the block, and the debit of the ticket issue is calculated by the booking clerk. The height of the cabinet is 4 ft. and depth 1 ft. 6 in. The 10-way machine is 2 ft. 4 in. wide and the larger machines vary according to capacity.

The Miniprinter machine, whilst differing from the Rapidprinter in its internal mechanical operation, is essentially a smaller version of that machine. It was designed to meet the demands of stations where intensive bookings are encountered, but where the limited number of destinations falling within this scope does not economically justify installation of the 10-way Rapidprinter. The Miniprinter casework is now standardised at six-way capacity and any number of print units from one to six can be accommodated on a chassis within it. Interchangeability of print blocks is also a feature of the Miniprinter.

Both machines have enabled the speed of ticket issue to be stepped up, and the resultant elimination of pre-printed stock enables saving to be made in storage space, and the time spent on stock records, requisitions and ticket checking. The cost of ticket printing is also reduced, and greater security is obtained by the substitution of blank card (which represents no debit, and is of little value if stolen) for orthodox printed tickets.

Handiprinter.

The Handiprinter, a Westinghouse Garrard unit, is to be tried experimentally by the Eastern Region. It is a free standing counter model machine measuring approximately 1 ft. by 10 in. by 1 ft. and is designed to issue hand-written tickets to those destinations which, because they are only infrequently asked for, cannot be obtained from the main machines in the ticket office. The presetting of levers on the machine enables the class, description, date, progressive number and fare chargeable to be printed on the paper ticket produced, leaving destination, route and validity to be inserted by hand afterwards. The Handiprinter contains an audit roll which records details of every ticket issued including particulars of each destination station. The latter is achieved by printing at the time of issue the station's code reference.

The audit roll thus contains information required for accountancy and statistical purposes. Also incorporated in the machine is a progressive cash total register, which enables the debit for any required period to be readily obtained. This machine, the first of its kind, is complementary to the Multiprinter and Flexiprinter, and whilst playing only a minor role in the working of the ticket office, it nevertheless achieves the important function of completing the mechanisation of all ticket issues.

Ultimatic ticket dispenser.

The Ultimatic ticket dispenser, a Bell Punch product, is eminently suitable for the rapid dating and issue of tickets to a limited number of destinations and is used mainly at suburban stations where the traffic is predominantly residential and is concentrated into peak periods. Each machine has five slots which are filled with packs of preprinted tickets, and in front of each slot is a depression lever. When this lever is operated, a ticket is automatically dated and ejected, and it is then only necessary for the clerk to tear off the ticket against a cutting edge and hand it to the passenger.

Although there is a separate dating mechanism for each slot, date changing is a simple operation requiring only a few seconds daily. All that is required is a slight turn of the knob controlling the dating spindle which adjusts the date simultaneously for each of the five slots. Although only five issues can be accommodated at any one time, tickets can be changed to meet the varying demands at different times of the day. Issues are accounted for by the booking clerk in the normal way. The machines have proved able to speed up the flow of traffic and reduce congestion at peak periods.

No fewer than 650 Ultimatics, made by the Bell Punch Co., Limited, 39, St James's Street, London, S. W. 1, have been, or are shortly due to be, installed in 500 stations throughout Great Britain. The majority of booking offices have only one machine, but large junction and termini stations have many more, such as the stations at Liverpool with 21, London Victoria with 20, and the recently opened booking office at Glasgow Central with four or Wimbledon, a London suburban station, with seven. Since this machine is but a development from the Ultimate machine used by many bus companies throughout the world, foreign visitors may well regard British Railways installations as a happy reminder of home.

Bellmatic dispensers.

Another unit by the Bell Punch Co., Limited, already employed on a large scale, which figured in the exhibition, is the Bellmatic ticket container and dispenser, which comprises tiers of ticket container units assembled in metal casework. Three types of casework, with varying capacities, are available to meet the individual requirements of stations where the traditional method of ticket issue exists. The contained units, which are interchangeable, are of steel construction with chromium-plated thumb pieces. The thumb piece itself is part of a simple ejection device, and when depressed it presents a preprinted ticket instantaneously into position between the operator's thumb and finger. A small spring clip is provided to hold any half tickets which may remain unsold and a slot on each unit indicates when stock is falling low. Identification of the tickets in each unit is provided in the form of a transparent celluloid cover beneath which is inserted a card showing the relevant particulars.

Setright long-range unit.

The Setright ticket register shown at Battersea was a portable machine widely used on road services. It has now been adapted for use by British Railways to operate on certain rural branch lines where tickets are required to be issued on the train. The machine is light in weight and is carried by the operator by means of a shoulder strap. As readers will be aware, tickets are printed and issued after wheels on the top of the machine have been preset to print the variable information required. These wheels record the day, month, fare stage number, up to seven different descriptions of ticket, and the fare from 1/2 d. to 19 s. 11 1/2 d. in graduations of one halfpenny.

In addition to recording the total issues and total cash debit, separate registers can be geared to record the total issues from any two descriptions of ticket. A statistical counter can also be set to record either the number of tickets issued of any particular value, or the number issued of any particular type, over a given period, and this can be re-set to provide statistics covering the whole fare range over a period of time. The majority of tickets are printed from a roll of paper (sufficient for over 500 tickets) inserted into the machine, but, if necessary, details can also be printed by the machine on to a hand-inserted card ticket. The internal ink ribbon is automatically fed through the machine and prints approximately 50 000 tickets. It can, if desired, be rewound and used once again. These machines are supplied by Setright Registers, Limited, of Hackney Wick, London, E. 9.

NEW BOOKS AND PUBLICATIONS.

[385 (08 (54)]

Report by the Railway Board on Indian Railways for 1955-1956. — Two volumes (8 1/4 × 13 in.) of 140 and 300 pages respectively, illustrated, numerous tables and maps. — 1957, published by the Director of Publications at New Delhi, edited by the Government of India Press, Calcutta.

The « Indian Railways » are in a period of a great expansion of the national economy. This is the dominant impression left by the annual report for 1955/56 recently published by the Railway Board. The « Indian Railways » operate 34 182 miles of lines; in 1955-56, the gross receipts amounted to 3 159 millions of rupees (1) and the expenditure to 2 588 millions; with the help of staff of 1 029 000 persons, it has been able to provide means for handling 38 773 millions of passenger miles and 36 420 million ton miles.

These few figures show relatively to other railway systems the importance of the Indian Railways. To form an idea however of the problems which present themselves and the magnitude of the task put forward by the Railway Board, it is sufficient to know that the second five year plan of the development of the governmental railways of India has just been started, and the amount overall (reduced) is 11 250 millions of rupees; its object is to increase the capacity for carrying passengers by 30 % and goods by 66 % (the expenditure of the first five years plan amounted to 4 240 millions of rupees).

Railway activity therefore is intense in India. It affects all fields : last year saw the approval for the construction of nearly 2 000 miles of new lines (20 % of metre gauge, the remainder standardised gauge); an important extension to the Locomotive Building Works at Chittaranjan and the opening, on the 1st October 1955, of the Integral Coach Factory at Perambur (which should be producing 350 coaches a year by 1959); the construction of a bridge 6 074 ft long over the Gange at Mokameh; the consolidation of tracks, superstructure and buildings, increase in the numbers of rolling stock, the acceleration of trains (which had gone back slightly during the year), increase in the use of railway lines, the electrification of the large part (2).

The report refers to many other problems; we may mention in passing : the making good the damage to the railway lines through violent storms, the investigation into the most economical strength of the permanent way gangs : the issue of regulations in the new common travel language Hindi; the reduction of the number of classes of carriages but the increased frequentation of air conditioned vehicles, the organisation of an important research and standardisation service, etc.

If the economy of the whole country is being developed, the part of the essential tool for India constituted by its railways is showing itself to be in full expansion. In this effort, the national industry plays a large part, about 75 % of the equipment supplied coming from this industry itself.

P. Sch.

(1) Rupee is equivalent to one shilling and six pence in British money.

(2) The Indian Railways have just put into use the 25 kV 50 Hz current.

[385 (09 .3 (485)]

Sveriges Järnvägar Hundra År *(The Centenary of the Swedish Railways).* — One bound volume (7 × 10 5/8 in.) of 664 pages, with many illustrations in black and white and colour and maps. — 1956, published by the Royal Administration of the Swedish State Railways.

The above is the title of a large work of more than 650 pages by the Royal Administration of the Swedish State Railways, which has just been published to commemorate the running of the first trains a centenary ago (1st March and 1st December 1856) over the private lines and those belonging to the State.

We get a firm idea of the sound way in which the book has been compiled from the first chapter. The Director General E. UPMARK deals in it with the present situation of the State system which during its own development has absorbed a large part of the private railways systems and which in its efforts to meet the multiple needs of the Swedish economy at the present time has reached the length of 15 036 km (9 350 miles) for a country which measures 1 700 km (1 060 miles) north to south. Dealing one after another with the problems before the Swedish Railways, he also sketches in his views of the future of the S. J.

Two long chapters are devoted by Mr. Arne SJÖBERG to the eminent place the Swedish Railways (S. J.) occupy in the economic life of the country from the beginning of the large development of building of railways from 1875 to 1890 during which period some 400 km (250 miles) of lines were opened to traffic each year and to the economic problems and those relating to rates which now as before and in particular during the very severe crisis 1918-1920, had to be dealt with by the Swedish Railways. The history of the taking over of the private lines by the State principally between 1915 and 1950 is dealt with by Mr. Erik MALMKVIST.

The problems relating to the location of the track, construction of the permanent way, the bridges and buildings, and those relating to the stabilisation of the ground are dealt with by Mr. John BJÖRK. He points out, in particular, the new lay-out of the lines in Stockholm Central, now being progressively completed without diminishing the number of trains; the doubling of lines in hand (it is expected to complete the work on the Göteborg-Stockholm line in 1958 and on the Malmö-Stockholm in 1960); re-inforced concrete sleepers with an original Swedish design of fastening, etc.

An important chapter written by Mr. Axel ALSTRÖM deals with the rolling stock. Among the remarkable points should be noted the thermic locomotive of 1 300 HP in which the heavy oil engine is used to generate hot gasses which are then passed through a turbine driving a jackshaft through reduction gearing; the railcars in which each of the 2 axles of the bogie are each fitted with a differential, the connection to the motor being made through a further differential, the whole ensuring running little affected by hunting movement; the electric locomotives with various designs of springing; the various problems in connection with special wagons which take advantage of the large Swedish Railway stock gauge; the sleeping cars fitted with all details which are so important from the passenger's point of view and in particular the special compartments for mothers with children of early age, etc. The chapter gives information on the arrangements made for the driving staff, a comparative study of the costs of traction of various types, and ends with information on the ferries, the latest of which is under construction and will be completed in 1958.

Mr. Th. THELANDER then takes up with his well known competence the whole aspects of the electrification of the railways and in particular those carried out before 1931, the year in which the first

investigation was carried out on the occasion of the 75th Anniversary of the Swedish State Railways. During the last 25 years the total length of the electric lines has increased from 910 (560 miles) to 6 100 km (3 800 miles). This was not done without difficulty! The Author begins by calling to notice the principal characteristics of the system of supply (16 kV, 16 2/3 Hz) starting from the 3 phase high tension grid through single-3-phase rotary convertors carried on wagons so equipped as to allow the electric power to be transferred in the two directions: this is followed by the return current lines which use systematically negative boosting transformers uniformly distributed over the system to reduce the induction effects; the method of constructing the overhead lines, the general organisation of the work of electrifying. He gives in detail the causes which lead to the development of the various types of locomotives of which the most recently supplied have a HP weight ratio of 54 t (type Ra, wheel arrangement Bo-Bo, 3 300 HP, 61 tons), which have made it possible to reduce the timings of the run Stockholm-Göteborg from 12 h in 1897 to 4 h 40 min in 1956. He ends by giving a very great amount of economic data.

Mr. Ernst JONSON, so far as he is concerned, has made a very thorough study of traffic currents and the tendencies which show themselves in this question, whereas Messrs. Henri KJELLVARD and Fred ARNELL deal respectively with the staff problems and the activity of the subsidiary S. J. Company the « Trafik-Restauranger » so well known to passengers.

The work ends with many maps which indicate to begin with the first projects for building railways (1840-1850) and subsequently for each period the regular and noticeable increase in this fine railway system now having happily reached its Centenary.

P. SCH.

[**625** .13 (44 + 45)]

Centenario del Traforo del Frejus *(Centenary of the piercing of the Frejus)*, 1857-1957. — A pamphlet (8 1/4 × 11 1/2 in.) of 34 pages with a plan attached and many illustrations. — Reprint of *Ingegneria Ferroviaria*, No. of July-August 1957. — 1957, Rome, General Management of the Italian State Railways, Piazza Croce Rossa.

The General Management of the Italian State Railways has just published a pamphlet devoted to the work of driving through the Mont Cenis tunnel in order to celebrate its centenary.

This pamphlet contains two articles, one written by Mr. Livio JANNATTONI dealing with the historical aspect of the tunnel and the other due to Dr.-Ing. Ferruccio PISANO for the technical aspect, the whole taken from *Ingegneria Ferroviaria*, Nos. 7 and 8 of July-August 1957.

We will briefly consider the historic part in which the author brings out the role played by Giuseppe Francesco MÉDAIL (1784-1844), who was not a scientist but a modest road contractor.

MÉDAIL sent to the Government his report in which he expressed himself in particular as follows: « It would be well to improve the road from Turin to Chambéry. For this purpose, it is desirable to abandon the present Mont Cenis road after having tunnelled the Alps at the shortest point which is under the Frejus Mountain, situated between Bardounèche and Modane. »

The historic part is freely illustrated and includes, besides the ends of the tunnel, certain works in connection with driving the gallery and the statues of the principal engineers who completed the work and in particular that of SOMMEILLER erected in his memory at Modane Station.

It is necessary to remember that for this first tunnel driven through the Alps the modern equipment was not available.

In addition to the geodetic operations and the study of the lay-out in plan and elevation, Dr.-Ing. PISANO details at length about the establishment of the Workshops at the two ends of the tunnel with the intervention of SOMMEILLER who, in collaboration with the Belgian Company Messrs. Cockerill succeeded in solving the question of the use of compressed air and the SOMMEILLER drills which rendered the greatest service in completing the first tunnel driven through the Alps.

The pamphlet ends with a very full table dealing with the traffics using the Frejus and the Simplon tunnels.

J. D.

[**385** (061 .4 (73)]

Association of American Railroads. — Signal Section. — Committee Reports, 1956 fiscal year (September 22, 1956 - September 21, 1957). — One volume of 205 pages (6 × 9 7/8 in.), with figures and diagrams, published by the A.A.R., Signal Section, 59 East Van Buren Street, Chicago 5, Illinois.

This brochure contains the reports discussed at the September 1957 Session on the occasion of the 58th Meeting of the Signal Section.

The various reports of the 12 Committees, the titles of which are given below are published therein.

Economics of railway signalling (heating points and crossings, track brakes, standardisation, C. T. C., etc.);

Signal shop practice (Ohm meter for testing rail joints);

Automatic block signalling (effect of long welded rails on the insulated joints, equipment to avoid sparks when transferring combustible liquids or gases, standardisation of protective devices);

Specifications (relating to direct current and alternating current relays, instruments for locating defects in cables);

Standardisation and investigation into materials (incandescence lamps, coloured screens, lead batteries, dry batteries, protection at level crossings, insulated cables using polyvinyle);

Electronics (application of transistors, recommendations relating to telecommunications by carrying currents, selection of frequency voltmeters, testing of electronic valves);

Standardisation of circuits (automatic blocks, interlocking, signalling of shunting yards, etc.).

F. B.

[**625** .28 (931)]

PALMER (A.N.) and STEWART (W. W.). — **Calvalcade of New Zealand Locomotives.** — One bound volume (7 1/2 × 9 7/8 in.) of 144 pages, with 2 maps and many illustrations. — 1956, Wellington : A. H. & A. W. Reed, Publishers of New Zealand Books, 182, Wakefield Street. (Price : 25 shillings).

« Calvalcade of New Zealand Locomotives » retraces the evolution of locomotives in New Zealand from the putting into service in 1863 of the first line of 3.5 miles up to the present day when the Government railway system exceeds 3 400 miles.

The various stages of this remarkable development which after overcoming very many vicissitudes travel from the 2-4-0 primitive locomotives of the Canterbury Railway of 1863, to the powerful 4-8-2 class Ja steam locomotives of 1946, to the BoBoBo electric locomotives class Ew of 1951 and the A1A-A1A Diesel locomotives class Dg

of 1956, will be followed with the liveliest interest.

A separate chapter is devoted to each of these stages. In each of these Messrs. PALMER and STEWART, after a short introductory note which re-creates for the reader the atmosphere of the time give particulars in turn of the different types of locomotives put into service. These chapters, which give the principal characteristics of the locomotives and the report of the outstanding facts in their history are all illustrated by photographs taken from the archives of the period.

Written in a breezy style and not without humour the work by Messrs PALMER and STEWART will awaken without doubt by the complete and frequently not previously published documentation it contains, the greatest interest amongst those who are enthralled by the history of railways.

R.S.

[**656** .257]

Dr.-Ing. habil. Wilhem SCHMITZ. — **Gleisbilder.** *(Illuminated track diagrams and panels)*. — One volume of 160 pages (8 × 5 3/4 in.) copiously illustrated. — Frankfort : Dr. Arthur Tetzlaff Verlag, Niddastrasse, 64. (Price : DM 10.—.).

If the optical control boards used since as long ago as 1900 in the signalling boxes to show the state of occupation of the tracks, the position of the points and signals, were originally aids, they have gradually become one of the essential factors in the working of signal installations especially in up-to-date lay-outs with levers or miniature push-buttons. Their importance has become even greater in the cases which are very numerous in which the press buttons of sufficiently small size have been incorporated in the board, thereby converting this into a living diagram reproducing topographically the whole of the points and crossings in the stations and enabling it to be easier to serve wide areas.

The various progresses made in ease of operation, which have been very extensive in the last 15/20 years, have only been obtained as a result of a detailed technical investigation into the whole of the details of construction. The book « Gleisbilder » by Dr. Ing. habil. W. SCHMITZ has been devoted to this investigation. For three principal types in which the signalling installations were carried out namely the boards completely separated from the push buttons and levers, the boards which form an extension over a desk on which are arranged the small levers or the free levers of the miniature type and those in which the operating buttons are incorporated in the boards themselves (formation of routes by entering and leaving buttons), the author describes details of construction (lamps, light signals, illuminated lines, keys, buttons, counters, supporters, etc.) their dimensions and best arrangement, the method of showing the routes set up and occupied by trains, the means employed to indicate the lines closed (for works), the conditions of the signalling of the line or their working by neighbouring boxes or for giving orders for setting up paths (stations with control boxes).

The very complete examination of the constructive solutions given to these problems by the some 25 to 30 principal European or American Builders ends by the description of typical tables for traffic controllers and by many examples of installations carried out.

The book illustrated by many photographs will certainly interest all those who wish to document themselves on the various types of modern signalling boxes that have been built.

P. Sch.

CORRIGENDUM.

BULLETIN FOR THE MONTH OF JUNE 1957.

Article : **The railways of Western Europe and America and their economic development,** by Dr. Robert KALT.

In the analysis of the evolution of the traffic during the years 1954 and 1937/1938, it has been unfortunately omitted to take into account that in Sweden, several private railways have been taken over by the State during the period considered for the comparison.

This naturally affects particularly the basis of the kilometric calculation of the system.

Therefore, the following alterations must be made to the particulars given for SWEDEN :

TABLE 4 (p. 498) :	*1954*	*Index* 37/38 = 100
Train-km (millions)	135.0	142
Passengers (millions)	124.0	165
Tonnes (millions)	40.1	118

TABLE 5 (p. 499) :	*1954*	*1937/1938*
Transport distance per inhabitant . . .	848	505
Passenger km per seat (thousands) . . .	23.5	16.2
T-km per t capacity (thousands)	8.6	6.5
Freight traffic receipts (passenger traffic = 100)	186	186

TABLE 6 (p. 501) :	*1953*	*1958*	*Index*
	(as a % of the national income)		38 = 100
Passenger traffic receipts	1.12	1.13	99
Freight traffic receipts	2.09	2.05	102

TABLE 7 (p. 502) :	*1953/54*	*Index* 37/38 = 100
Pas.-km (thousand million)	6.1	193
T-km (thousand million)	9.1	169

R. K.

MONTHLY BIBLIOGRAPHY OF RAILWAYS (1)

PUBLISHED UNDER THE SUPERVISION OF

P. GHILAIN,

General Secretary of the Permanent Commission of the International Railway Congress Association.

(JANUARY 1958)

[016. 385 (02]

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In French.

1957 697
MISSENARD (H.).
Cours supérieur de chauffage, ventilation et conditionnement de l'air. Livre I : **Principe et description des installations.** 5e édition entièrement refondue.
Paris, Eyrolles, éditeur. Un volume (16 × 25 cm), de 386 pages avec 221 figures et 3 planches hors-texte. (Prix : 3 300 fr. fr.)

1957 621 .3 (02
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Electrotechnique Générale. 5e édition.
Paris, Dunod, éditeur. Un volume (10 × 15 cm). (Prix : relié, 480 fr. fr.)

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Paris, Librairie Dunod. Un volume (15.5 × 23.5 cm), de 328 pages, avec 216 figures. (Prix : 2 800 fr. fr.)

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Schrumpfungen, Spannungen und Risse beim Schweissen.
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The measurement of soil properties in the triaxial test.
London : Edward Arnold, Ltd., 41, Maddox Street, W. 1. (Price : 70 s.)

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BOGUSLAVSKY (B.W.).
Design of reinforced concrete.
London : The Macmillan Company, 10, South Audley Street, W. 1. (Price : 42 s.)

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Famous locomotives of the world.
One volume (7 1/2 × 5 in.) of 143 pages, illustrated.
London : Fredk. Muller Limited, Ludgate House, 110, Fleet Street, E.C. 4. (Price : 9 s. 6 d.)

1957 621 .131 .3 (42)
HOLCROFT (H.).
An outline of Great Western locomotive practice.
Hampton Court, Surrey : Locomotive Publishing Company, Limited, Craven House, The Green. (Price : 25 s.)

(1) The numbers placed over the title of each book are those of the decimal classification proposed by the Railway Congress conjointly with the Office Bibliographique International, of Brussels, (See « Bibliographical Decimal Classification as applied to Railway Science », by L. WEISSENBRUCH in the number for November 1897, of the *Bulletin of the International Railway Congress*, p. 1509).

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High-speed Diesel engines.
London : Chapman & Hall Ltd.
One volume of 578 pages, illustrated. (Price : 65 s.)

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Gas turbine materials. A survey of high-temperature materials and their application to the gas turbine.
London : The Temple Press Limited, Bowling Green-lane, E.C.1. (Price : 25 s.)

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Revised and edited by H.C. Casserley.
London : Frederick Warne & Co. Ltd., Chandos House, Bedford Court, Bedford Street, W.C. 2.
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[016. 385 (05]

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FRANÇOIS (M.). — **Barres longues sur longrines Laval.** (600 mots & fig.)

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Electronics in a U.S.A. marshalling yard. (1 100 words & figs.)

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The Railway Gazette, September 20, p. 333.
High speed junctions and crossovers. (800 words & figs.)

1957 625 .232 (54)
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BHATTA CHARAYA (A.C.). — **Indian Railways third class sleeping coaches.** (700 words & figs.)

1957 621 .33 (42)
The Railway Gazette, Sept. 27, p. 363; Oct. 4, p. 393.
Design and construction in main-line electrification (900 words & figs.)

1957 621 .338 (62)
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Electric trains for Cairo-Helwan service. (300 words & figs.)

1957 **656** (42)
The Railway Gazette, Oct. 4, p. 390; Oct. 25, p. 473.
Ten years of nationalised transport-2-3. (3 000 words.)

1957 **625** .285 (42)
The Railway Gazette, October, 4, p. 396.
Diesel-electric sets for Hampshire lines. (200 words & figs.)

Diesel Railway Traction. (London.)

1957 **621** .431 .72
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Early pressure-charged units. (1 300 words & figs.)

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Diesel Railway Traction, September, p. 349.
The French TEE train sets. (3 600 words & figs.)

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VON KIENLIN (M.). — **The relations of engine and transmission.** (1 800 words & figs.)

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SHAIKH (W.A.). — **Diesel traction on Eastern Bengal Railway.** (3 500 words & figs.)

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ALLEN (C.J.). — **British locomotive practice and performance.** (1 900 words, tables & figs.)

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The Railway Magazine, November, p. 791.
Standard 35-ton tank wagons for British Railways. (500 words & figs.)

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The Railway Magazine, November, p. 793.
HARRISON (M.A.). — **Indian 4-6-0 locomotives.** (600 words & figs.)

1957 **656** .222 .5 (4)
The Railway Magazine, November, p. 799.
International passenger services in Eastern Europe. (800 words & figs.)

Railway Track and Structures. (Chicago.)

1957 **625** .13 .(73)
Railway Track and Structures, September, p. 47.
BENSON (G.W.). — **At this bridge install girders to widen roadway like this.** (300 words & figs.)

1957 **624** (73)
Railway Track and Structures, October, p. 41.
CHAPMAN (W.E.). — **Blueprint for total B & B mechanization.** (500 words & figs.)

1957 **625** .144 .4 (73)
Railway Track and Structures, October, p. 44.
Compressors now go on-track. (200 words & figs.)

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Railway Track and Structures, October, p. 48.
« Wire » lining 120-ft length « sights » for power track liner. (600 words & figs.)

In Spanish.

Ferrocarriles y Tranvias. (Madrid.)

1957 **624** (44)
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MENDIZABAL (D.). — El **nuevo puente colgado** sobre el rio Sena, en Tancarville. (500 palabras.)

1957 **385**
Ferrocarriles y Tranvias, vol. 24-I, nº 269, p. 5.
NAVARRO GIL (L.). — **Problemas ferroviarios.** (3 000 palabras.)

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Ferrocarriles y Tranvias, vol. 24-I, nº 269, p. 10.
DE LA FUENTE (J.). — Ferrocarriles americanos. Los **créditos para compras de material.** (2 000 palabras.)

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Ferrocarriles y Tranvias, vol. 24-II, nº 270, p. 30.
PEREZ-BROTONS (D.). — Los **nuevos coches de la Dirección General de Correos y Telecomunicación.** (1 500 palabras & fig.)

1957 **385** .587
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SANCHEZ PEREZ (J.). — El **factor humano en la productividad de los talleres ferroviarios.** (2 000 palabras & fig.)

In Italian.

Giornale del Genio Civile. (Roma.)

1957 **624** .9
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CERADINI (G.). — Alcune considerazioni sui **metodi di calcolo per iterazione dei telai.** (15 000 parole, tabelle & fig.)

1957 **624** .9
Giornale del Genio Civile, luglio-agosto, p. 616.
PAGANO (M.). — Il **calcolo dei telai a nodi spostabili** con l'introduzione di vincoli ausiliari interni. (600 parole & fig.)

Ingegneria Ferroviaria. (Roma.)

1957 **621** .331 (45)
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PROSPERI (L.). — **Automatismi e comandi a distanza nelle sottostazioni di conversione a c.c. 3 kV.** (3 500 parole & fig.)

1957 **656** .223 .1
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TILLI (F.). — Vecchie e nuove **modalità di scambio del materiale da viaggiatori** fra le ferrovie europee. (1 500 parole & tavole.)

1957 **625** .232 (45)
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FASOLI (M.). — Le **carrozze-cuccette** delle F.S. (2 000 parole & fig.)

La Ricerca Scientifica. (Roma.)

1957 **62** (01
La Ricerca Scientifica, settembre, p. 2701.
CASTIGLIONI (A.). — Alcune anomalie nel funzionamento degli **estensimetri elettrici a variazione di resistenza.** (2 000 parole & fig.)

Rivista di Ingegneria. (Milano.)

1957 **621** .3
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MASSA (E.). — L'assorbitore **dinamico non lineare non smorzato con vincolo elastico di rigidezza costante a tratti** *(continua)*. (2 500 parole & fig.)

In Netherlands.

De Ingenieur. (Den Haag.)

1957 **624** & **721** .1
De Ingenieur, nr 41, 11 october, p. B. 85.
Verslag over het IVe **Internationale Congres voor grondmechanica en fundeeringstechniek** te Londen (12-14 augustus 1957). (4 000 woorden.)

1957 **537** .7
De Ingenieur, nr 42, 18 october, p. E. 119.
KRAUSHAAR (E.). — Die neue Entwicklung der **Fernmesstechnik.** (4 000 woorden & fig.)

1957 **656** .254
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Radiotelefoonverbinding met en in treinen. (200 woorden & fig.)

Spoor- en Tramwegen. (Den Haag.)

1957 **656** .224
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HARMSEN (B.T.). — De **spoorwegen voor het postvervoer.** (2 000 woorden & fig.)

1957 **625** .13 (492)
Spoor-en Tramwegen, nr 21, 10 october, p. 330 en p. 333.
De **Velsertunnel** officieel geopend. (2 000 woorden & fig.)
De **verlichting van de Velsertunnel.** Gemakkelijke aanpassing der ogen mogelijk gemaakt. (1 000 woorden & fig.)

1957 **656** .254
Spoor- en Tramwegen, nr 21, 10 october, p. 336.
Het **overwegvraagstuk** thans in de bioscoop. (800 woorden & fig.)

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Spoor- en Tramwegen, nr 22, 24 october, p. 341.
MOOLHUIJZEN (Chr.). — De **dieselelektrische locomotieven serie 450** der N.S. (1 600 woorden & fig.)

1957 **385** .517 .1 (492)
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LANDSKROON (F.P.A.). — De **pensioenen bij de spoorwegen.** (2 500 woorden.)

In Serbo-croate (= 91.882).

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JAGODIČ (F.). — **Problem of the system of the future electrification of the Yugo-Slav Railways.** (2 500 words.)

In Czech. (= 91.886.)

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1957 **624** (437) = 91 .886
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MOZIS (V.). — **Three schemes for a large prestressed concrete bridge (326 m).** (6 000 words, tables & figs.)

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VIHAN (J.). — **Reconditioning of a railway masonry viaduct.** (2 000 words & fig.)

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MACHAC (J.). — Present acquired **experience with concrete sleepers** on the Czcheco-Slovak Railways. (1 200 words & figs.)

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KÚKOL (J.). — Development of the installation for **supplying 300 V D-C current on the electrified lines** of the Czcheco-Slovak Railways .(3 000 words & figs.)

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NEJEPSA (R.). — **Duration of the economic utilization of the (motor) locomotive.** (1 500 words & figs.)

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